HVAC System Energy Consumption Reductions and Energy Efficient Design Strategies for the Heating and Cooling of Buildings

The 12 MPG Prius

or “It All Comes Down To How You Control It”

Net Zero Energy Buildings
USACE ERDC CERL
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Overview

- **Making relatively minor adjustments to new and retrofit HVAC system designs can cost effectively cut energy consumption by over 50% when compared to "normal" energy efficient design strategies.**
- **As important (or perhaps more important) as the equipment selections and mechanical design strategies, are the methods used to control the equipment.**

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Overview, Continued

- **Merely having great equipment and a fantastic system design does not guarantee that a system will be energy efficient.**
- **Typical "energy efficient" control strategies leave a substantial amount of savings on the table.**
- **Control system strategies that cannot be easily understood by the operating staff will quickly be disabled, and the system will revert to manual operation, and savings will be lost.**

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Summary

- To reduce energy consumption, you need to minimize system losses:
  - **Utilize larger ductwork and air distribution equipment to reduce airside pressure drops.**
  - **Utilize large, low pressure drop sound traps where sound traps are required.**
  - **Utilize larger piping and 45° angle piping connections to minimize piping system losses.**
  - **Utilize 1170 RPM pumps and low RPM fans to improve efficiency.**
  - **Utilize "Variable Speed Everything" Design Strategies**

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Summary

- **To reduce energy consumption, you need to improve heat transfer efficiency:**
  - **Utilize large face area cooling and heating/reheat coils to maximize system temperature differentials and minimize air and water pressure losses.**
  - **Large cooling coils reduce air pressure drops and allow the use of high chilled water supply temperatures, while still dehumidifying, raising chiller system efficiency.**
  - **Large heating and/or reheat coils reduce air pressure drops and allow the use of low hot water supply temperatures. This can allow the effective use of low quality reclaimed heat, and/or highly efficient condensing boilers.**

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Summary

- **To reduce energy consumption, you need to improve heat transfer efficiency:**
  - **Install the supply fan upstream of the cooling coils (a “blow-thru” coil configuration) to maximize system efficiency.**
  - **Utilize large chiller shells and the maximum tube counts to reduce the refrigerant to water approach temperatures.**
  - **Utilize large surface area cooling towers, or wet/dry coolers with close approach temperatures to the wet bulb temperature.**

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Summary

- **Utilize reclaimed energy whenever possible.**
  - **Typical moisture control designs require a substantial amount of new cooling and re-heat energy to dry the air and reduce relative humidity.**
  - **Innovative dehumidification and reheat systems can reduce energy waste by 50% to 70%.**
  - **Air to air heat exchangers may be cost effective in some situations and climates.**

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Summary

- **Poor control strategies can waste more energy than great equipment can save.**
- **The controls must adapt to changing end use loads and conditions and allow the operating staff the ability to tune the system to meet the needs of the various connected loads.**

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Summary

- Energy Conservation = water conservation.
  - If a cooling system is water cooled, saving energy will reduce the amount of heat that has to be rejected by evaporating water, so water savings will result.
- It may be possible to capture and re-use condensed moisture off of the cooling coils.

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Moisture Control

- In order to reduce the potential for biological issues, the building envelope and HVAC system must be designed in concert to properly control moisture.
- Relative Humidity in the facility must be maintained within acceptable boundaries to promote occupant comfort and health, while reducing the potential for biological issues.

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Save Energy, Improve Comfort

- It is possible to save a substantial amount of energy, while at the same time improving the comfort within a facility.

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E-mail from Dave Manley – 71% HVAC System Energy Reduction

- Gentlemen,
- **An astonishing 70.97% reduction in HVAC usage** with the new Chiller running 49 more hours versus last year! Overall usage down a remarkable 33.89%! Small increase in Common Area usage due to increase in number of RMU's and TI's. Cooling Tower water usage up 14% due to Duncanization of Cond. Water, reducing Cond. Water set point to help lower chiller KW. Another good month at Inland Center!
- **David Manley | Operations Manager**
- **Inland Center Mall**
- **500 Inland Center**
- **San Bernardino, CA, 92408**

Note – the entire HVAC system, including the chillers, the chilled water pumps, the cooling towers and the supply and return fans are on a separate utility meter, so year to year comparisons are made easy. This is not weather corrected data, but raw utility meter data.

E-mail from Dave Manley – 86% HVAC System Energy Reduction

Gentlemen,

- **If we reduce usage anymore…we can turn everything off and go home!** Overall usage down an amazing 55.89% from last year. **HVAC usage down 86.59%** from last year, triple checked figures. Duncanization didn't effect Tower water usage like it did last month. Total monthly breakdown usage shows HVAC = 22%, INTERIOR = 50%, EXTERIOR = 28%. Only increase was in #2 COM. AREA due to increase in RMU's and contractors using common area power during TI's.
- **David Manley | Operations Manager**
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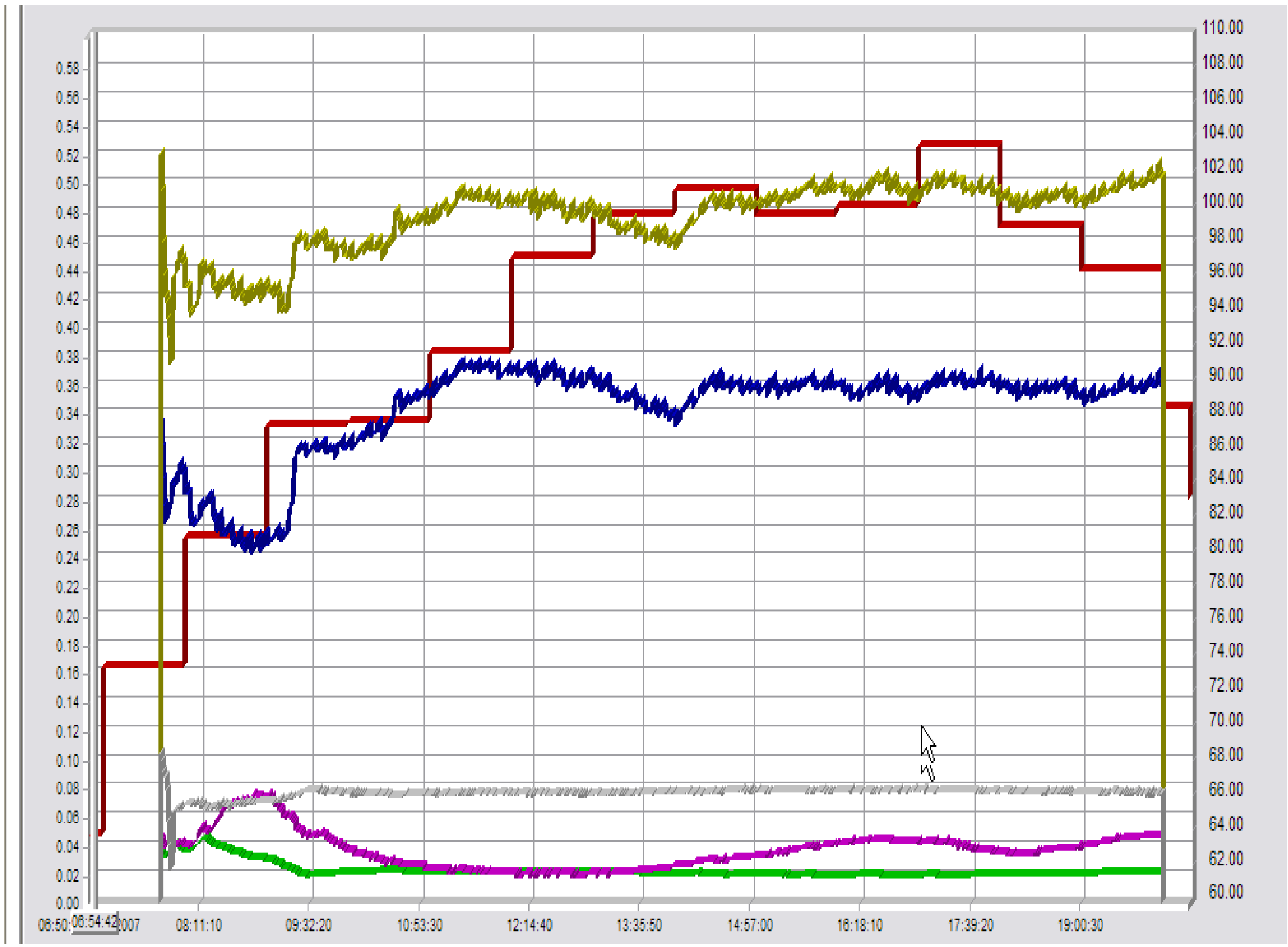
Sample VSE/LOBOS Design and Control Logic Results



6/30/07 Peak OSA Temperature = 103°F red line

Average TOTAL chiller plant efficiency = Less than 0.50 kW/ton yellow line

Chiller, Approx 0.35 kW/ton, blue line

Cooling Tower, Approx 0.08 kW/ton, white line

Chilled Water pump, Approx 0.04 kW/ton, pink line

Condenser water pump, Approx 0.03 kW/ton, green line

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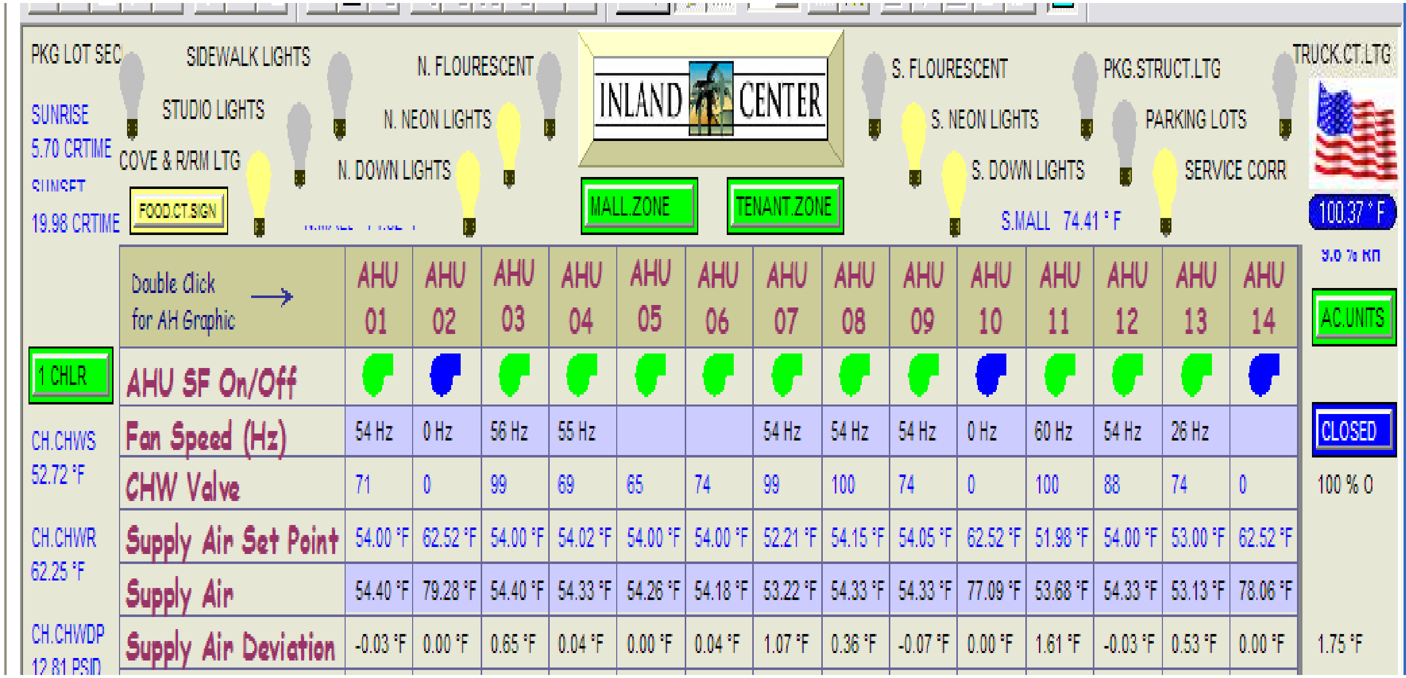
AHU Summary



100 degrees outside

52.7 °F CHW supply temperature

Max 54.5°F supply air temp. on all AHU's

At 100°F outside, we have a 1.8 degree approach temperature between the CHWS temp and the supply air temp leaving the coils. This allows very high CHW temperatures and thus very high system efficiencies. This compares to typical designs that use coils that provide a 10°F to 13°F approach temperature.

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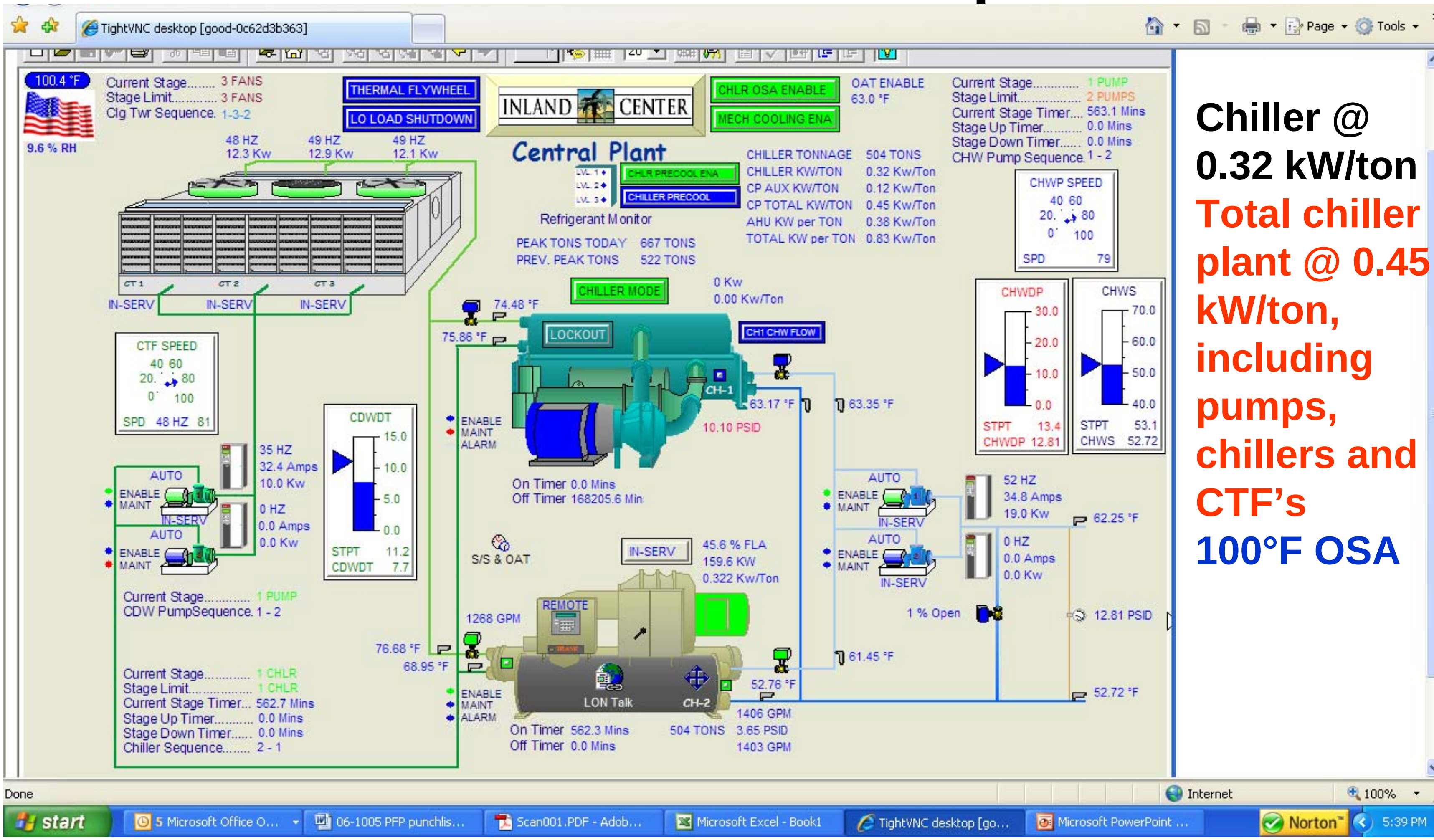
Chiller Plant Snapshot



Chiller @ 0.32 kW/ton

Total chiller plant @ 0.45 kW/ton, including pumps, chillers and CTF's

100°F OSA

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Complete day performance: tons and kW/ton 6-13-07

Blue = total plant (0.39 to 0.51 kW/ton), including all pumps, CH's and CT's

Yellow = load in tons (380 to 530 tons)

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Complete day performance: tons and kW/ton 6-14-07

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Complete day performance: tons and kW/ton 6-15-07

Blue = total plant (0.42 to 0.50 kW/ton), including all pumps, CH's and CT's

Yellow = load in tons (370 to 510 tons)

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Complete day performance: tons and kW/ton 6-16-07

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Complete day performance: tons and kW/ton 6-17-07

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Complete day performance: tons and kW/ton 6-18-07

Blue = total plant (0.41 to 0.48 kW/ton), including all pumps, CH's and CT's

Yellow = load in tons (380 to 490 tons)

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Complete day performance: tons and kW/ton 6-19-07

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Controls can Make or Break a System

- **If you controlled a car like the typical control strategy for an HVAC system, you would get crummy efficiency.**
- **Step on the throttle to get the engine capacity ready for the load, then step on the brakes to make the speed of the car match the desired speed.**

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Make or Break, continued

- **Pretty wasteful control strategies, but that is how most HVAC systems are controlled and operated.**
- **Fire up the chiller, drop the chilled water temperature, let the pumps run at a constant differential pressure setpoint, then use the AHU cooling coil control valves to waste all the differential pressure and control the excess capacity that was generated at the plant.**

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11.9 MPG on the Prius

by controlling it like HVAC Controls usually run

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Prius with “Efficient” HVAC Control Strategies

16 MPG – 26% less energy used to meet the same loads

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Prius with “Optimized” HVAC Control Strategies – 99 MPG

– 88% less energy used to meet the same loads

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Understanding the intent vs. being told “its too complex for you to understand”

- **The intent of the control routine must be easily understood by the least capable person that will be operating the system.**
- **If the intent of the control routines are not understood, the system will drop into manual mode in a hurry – everyone understands manual operation.**

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One Size Does Not Fit all

- **Buildings and HVAC system operate differently during startup than they do at 10:00 AM, noon, 3:00 PM or 7:00 at night.**
- **They operate differently on weekends than they do during the week.**
- **They operate differently during winter, spring, summer and fall.**
- **They operate differently when it is muggy than they do when Santa Ana winds are blowing.**
- **They respond differently when 1 floor is operating than they do when 24 floors are running.**
- **They respond differently when it is 65°F and sunny than they do when it is 65°F and foggy.**

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The Control Strategy Must Fit the Situation **(which changes continually)**

- A control system that treats all loads and situations the same is doomed to failure, or at least doomed to not save as much energy as it is possible to save.
- **Self Tuning Control Loops** will promote energy efficiency and Operating Staff confidence that they can leave the system alone.

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Control to the Loads

- **The system must respond to the needs of the loads, be it a single floor in the winter, or the entire facility in the heat of the summer.**
- **If the loads are not being properly cared for and met, the system is not doing its job.**
- **As soon as tenant complaints start coming in, the system will go to manual operation, if it is not easy for the Engineering Staff to tune and adjust.**

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The Basics Don't Change

- **To save substantial energy with any system, you have to incorporate central plant and airside optimization software.**
- **Central Plant Optimization** routines must look at the site loads – without knowing AHU load information, you cannot maximize energy savings, or respond to load changes properly.
- **AHU Optimization routines are required to minimize energy use and promote occupant comfort!**
- **The System Must be Operator Friendly** – If the person that designed it cannot explain it in an understandable manner to the people that operate the facility, it is too complex.

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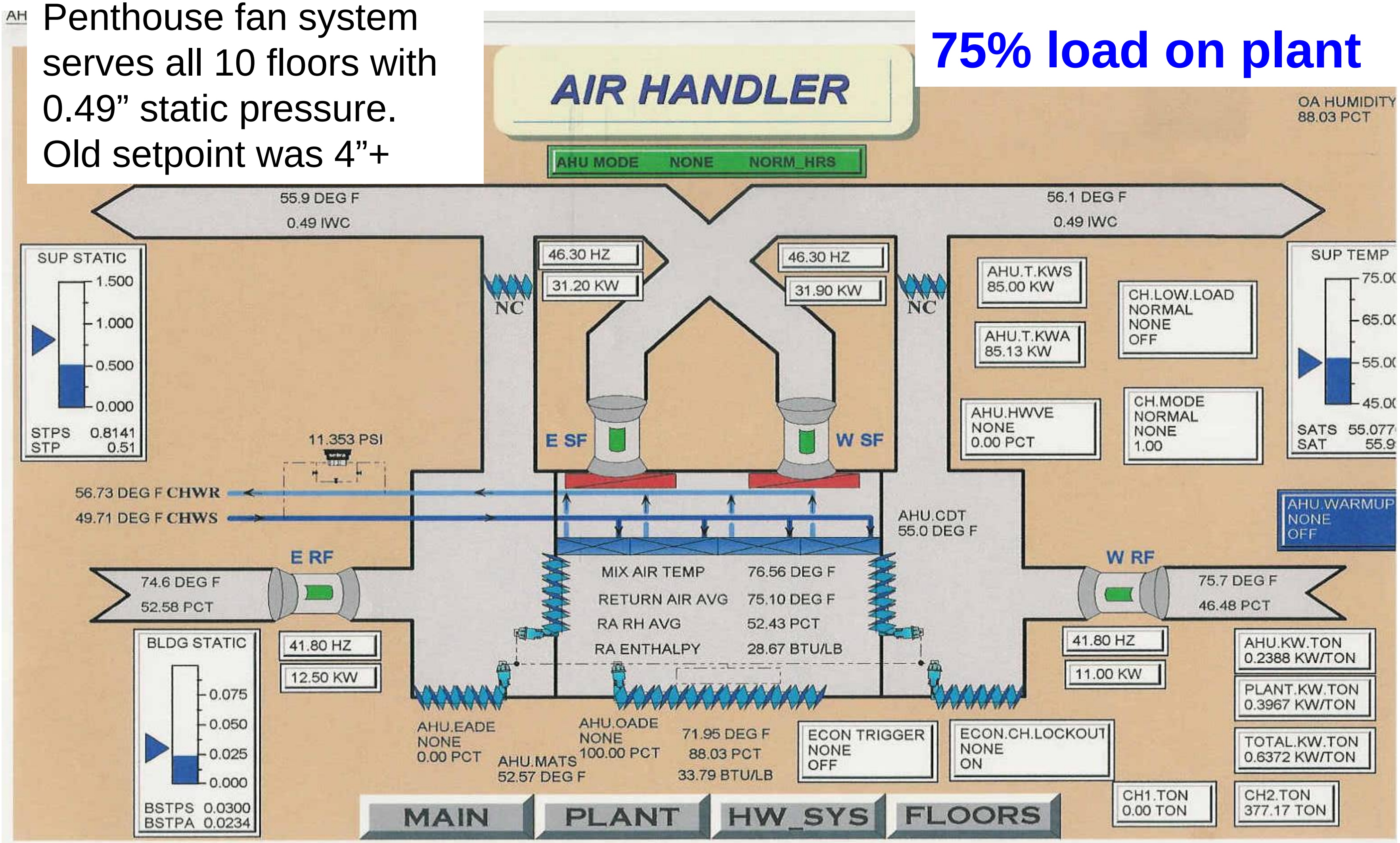
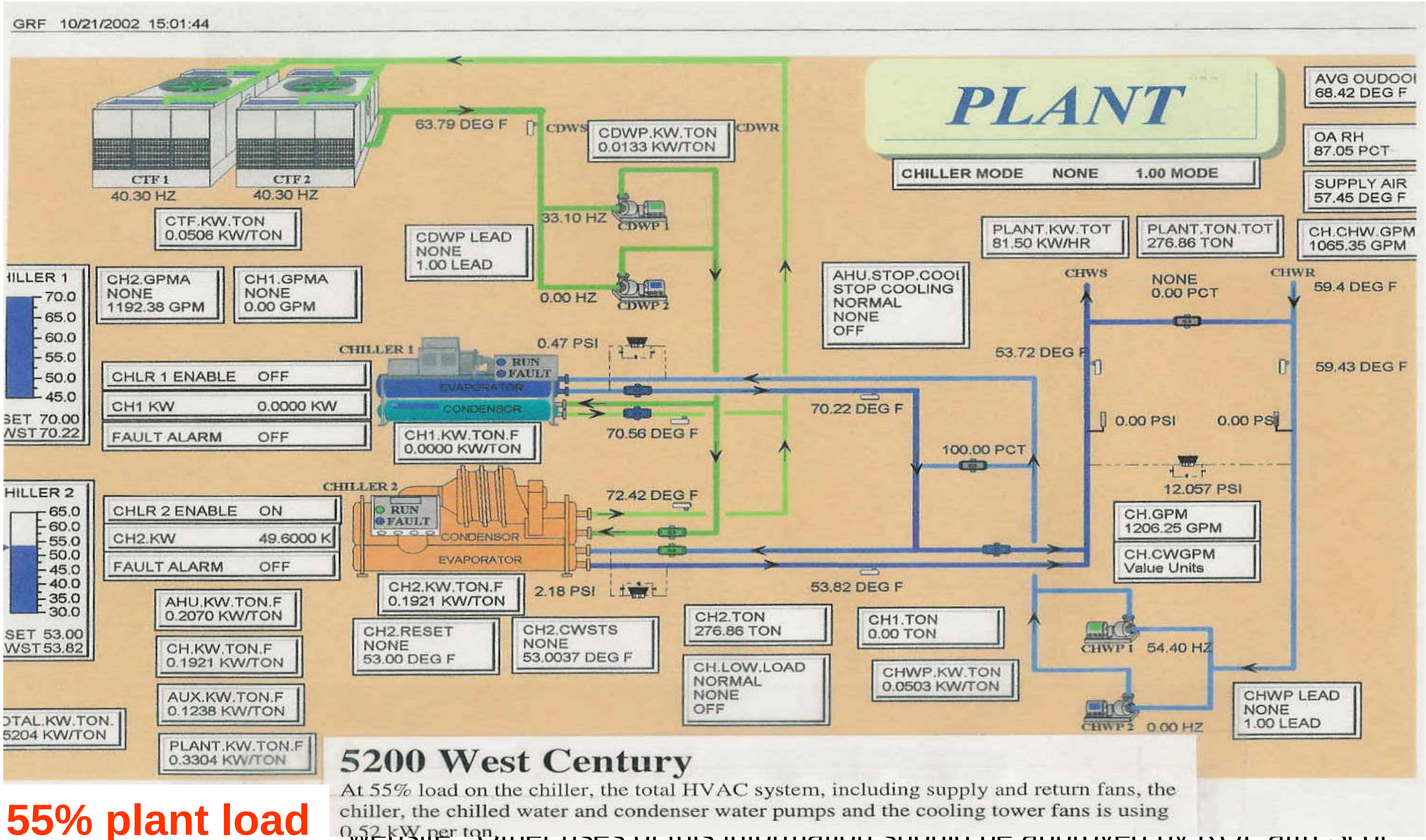
Load Based Optimization on AHU – 56°F at 0.49 INWC @ 10th Floor AHU@0.24 kW/T & **Total HVAC sys @ 0.64 kW/T**

Penthouse fan system serves all 10 floors with 0.49” static pressure. Old setpoint was 4”+

75% load on plant



Total Plant @ 0.33 kW/Ton and
Total HVAC System at 0.52 kW/Ton



55% plant load

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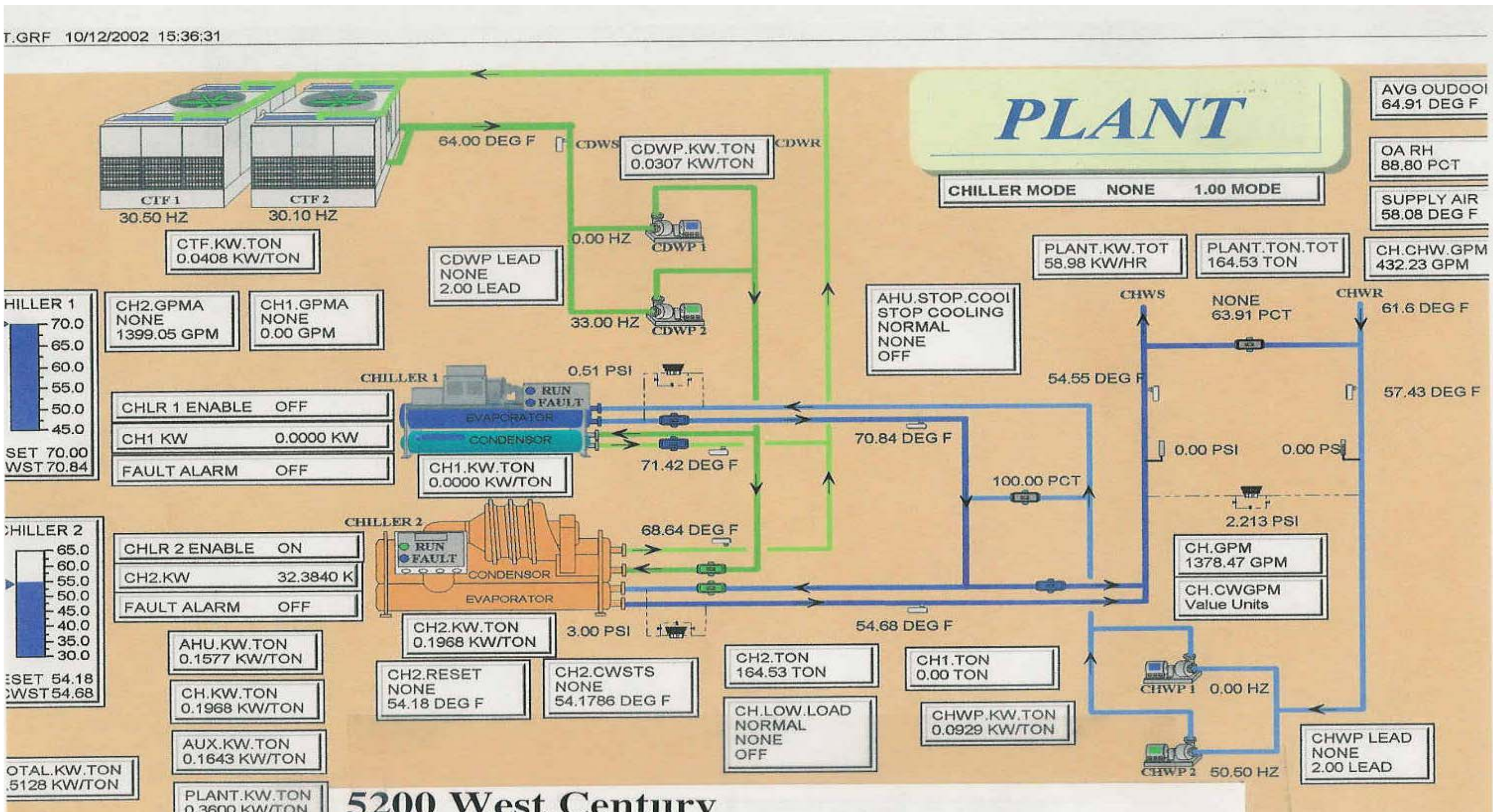
Total Plant Efficiency @ 0.38 kW/T – **Total HVAC System** (including Supply and Return Fans) @ 0.51 kW/Ton **(VSE/LOBOS)**



5200 West Century

At 33% load on the chiller, the total HVAC system, including supply and return fans, the chiller, the chilled water and condenser water pumps and the cooling tower fans is using 0.51 kW per ton.

33% plant load

Common Sense Control

- **The chiller plant should respond to the AHU loads, like a VAV box responds to the zone loads.**
- **If you controlled a VAV box independently of the load it serves, how many complaints do you think you would get each hour?**
- **How can you control a chiller plant independently of the loads it serves?**

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Load Based Optimization Saves "Invisible" Energy

- **If you run the chilled water too warm to try and make the chiller plant efficiency look good, you won't meet supply air temperature setpoints, causing the fans to speed up to deliver more air, since the VAV boxes will open up.**
- **This wastes a boatload of "invisible" fan energy. It is invisible, since many engineers ignore it, so it must not exist.**

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Operating Staff are the Key to Long Term Success

- **To ensure long term project success, the operating staff must understand the system, and they must buy into the design philosophy.**
- **We have all seen facilities with $500,000 DDC system time clocks because the operating staff was not made a part of the process, they did not understand how it was supposed to work, and they could not easily “tweak” the system to suit their tenant needs.**
- **Once a system goes to “man-u-matic” operation instead of automatic operation, savings go out the window.**

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Interim Summary

- **The LOad Based Optimizing System (LOBOS) AHU resets can dramatically reduce energy waste at the AHU's.**
- **Less waste at the AHU's results in less waste at the plant as well.**
- **Lower static pressures and higher supply air temperatures reduce energy waste and improve actual and perceived comfort.**
- **As loads vary, so do the static pressure and supply air temperature setpoints – higher loads equate to higher static setpoints and lower SAT setpoints.**

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Load Based Optimization is the Key to Project Success

- **If the central plant reset routines are not evaluating the loads at the AHU's and responding to those loads, you are doing one of two things:**
  - **you either waste central plant energy by running the chilled water too cold, or**
  - **you waste AHU energy by running the chilled water too warm, and the AHU's can't meet setpoint, ramping up as the static pressure drops off.**

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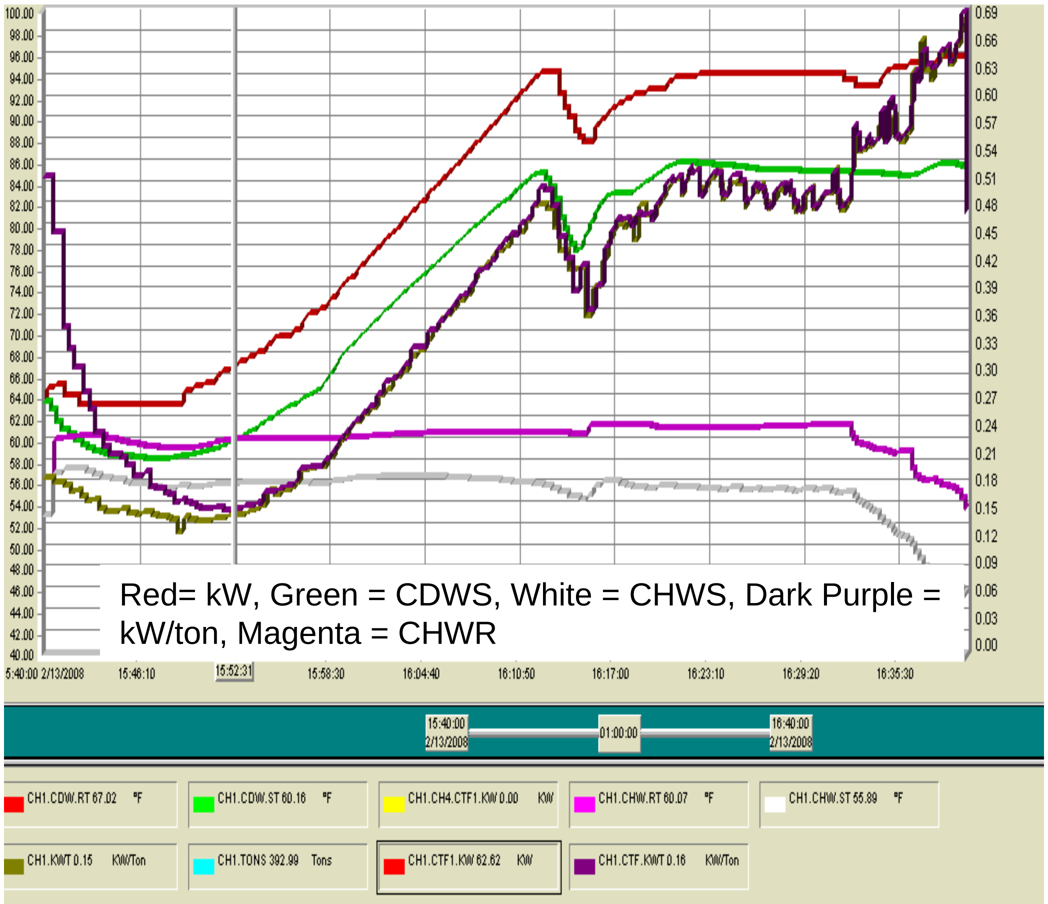
Effects of Improper System Control



Load Based Optimization System (LOBOS) controlling the chiller and cooling tower energy draw @ 0.16 kW/ton @ a 400 ton load.

Taking the system out of LOBOS control and raising the cooling tower temp to 80F and dropping the CHW temp to 46F push the energy draw to 0.69 kW/ton.

As CDWS temperature is increased to 80F, chiller + CTF energy increases by 300%. As CHW Supply temperature is dropped to 46F on top of the CDW increase, chiller + CTF energy increases by 425% +.

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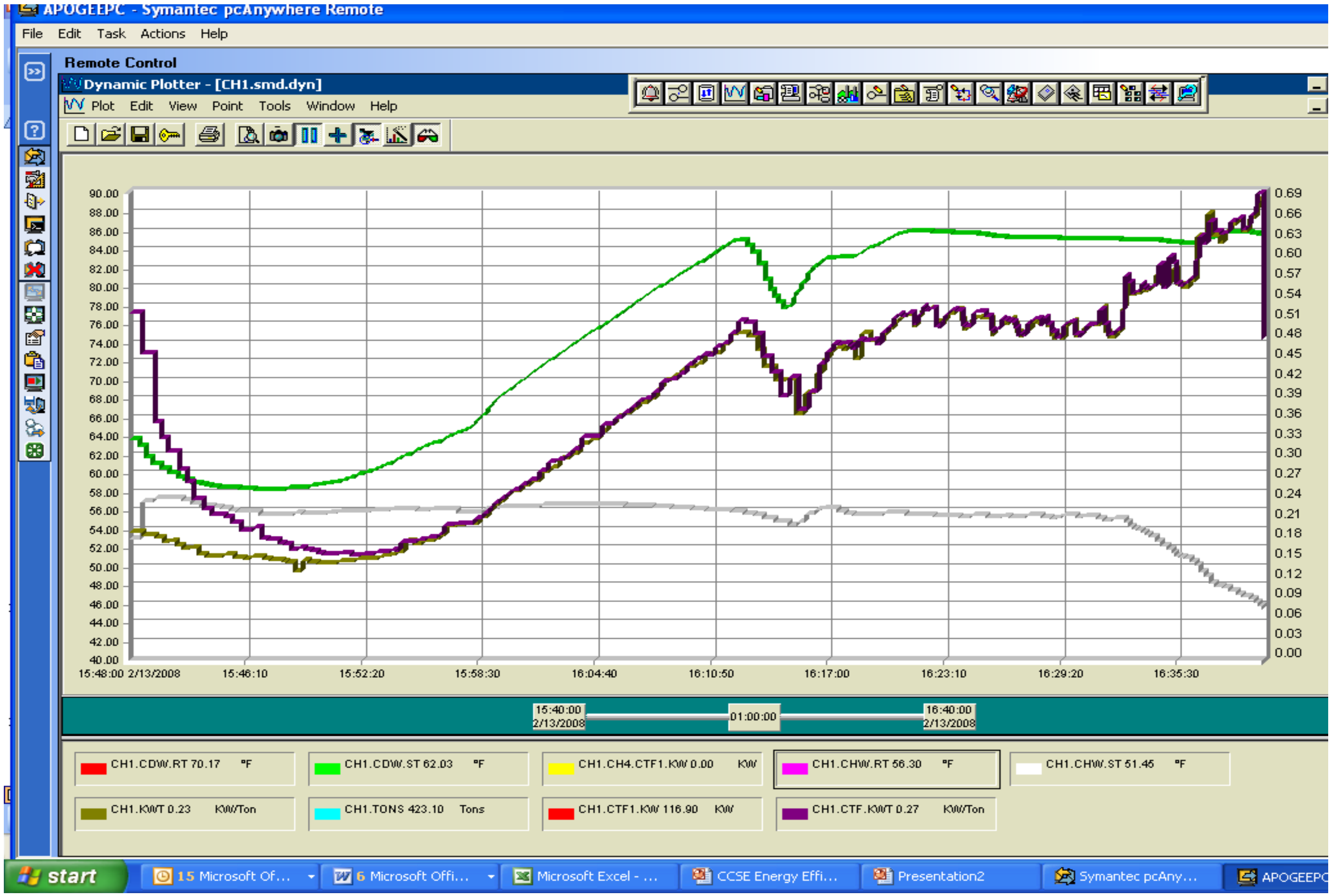
Refrigerant Lift vs. Efficiency



Greater lift (relative difference between CDWS and CHWS) equates to much worse system efficiency. 4 degrees of differential = 0.16 kW/ton, 40 degrees of differential = 0.69 kW/ton.

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Dehumidification/Reheat Systems

- **Many areas in the continental United States only have dehumidification problems in the summer.**
- **Other, more far flung locations require continual dehumidification and reheat to maintain comfort conditions and reduce biological issues.**

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Dehumidification Systems Problems

- **Billions of dollars in damage occurs each year due to biological growth caused by high Relative Humidity in facilities.**
- **This is preventable, and can be energy efficient at the same time!**

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Dehumidification Systems Problems

- **To reduce relative humidity in the spaces, the air must be dried out, typically by cooling it to 45°F to 55°F, then re-heating it to around 65°F typically by using electrical or hot water re-heat sources.**
  - **In this manner, very cold, 100% saturated supply air does not enter the spaces.**
  - **If cold, saturated air enters a room, biological issues will likely follow.**

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Dehumidification Systems Problems

- **High RH can be caused by the desire to save energy by reducing the dehumidification (cooling) and re-heat load on the system.**
  - **If you cool the air less, you save energy, but you increase the RH in the space.**
  - **If you dry the air out by cooling it to the correct level, but then do not re-heat the air (to save heating energy), the cold saturated air will encourage condensation everywhere it touches.**

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Dehumidification Systems Problems

- **At one site that we visited, biological growth issues created the need to completely re-build one of the living facilities. The new HVAC system uses chilled water for dehumidification and electric strip heaters (at 1.0 kW per room!) for reheat to lower the relative humidity of the spaces.**
- **While the design of the system will work, the annual energy consumption of this one 250 room facility will increase by approximately 1,500,000 kWh per year.**
  - **The strip reheat coils will soon be disconnected to try and comply with energy conservation goals, and the biological problems will return, creating a massive waste of funds, but coming out of a different, less obvious budget.**

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High Efficiency Dehumidification Systems (HEDS)

- **We've developed a high efficiency dehumidification system that can reduce energy consumption by 52% (high loads) to 72% (low loads), while improving chiller plant and boiler plant efficiency.**
- **Combining VSE design strategies with Load Based Optimization Controls, (LOBOS) and HEDS can reduce facility energy consumption, while reducing the RH to reduce the potential for biological damage to occur.**

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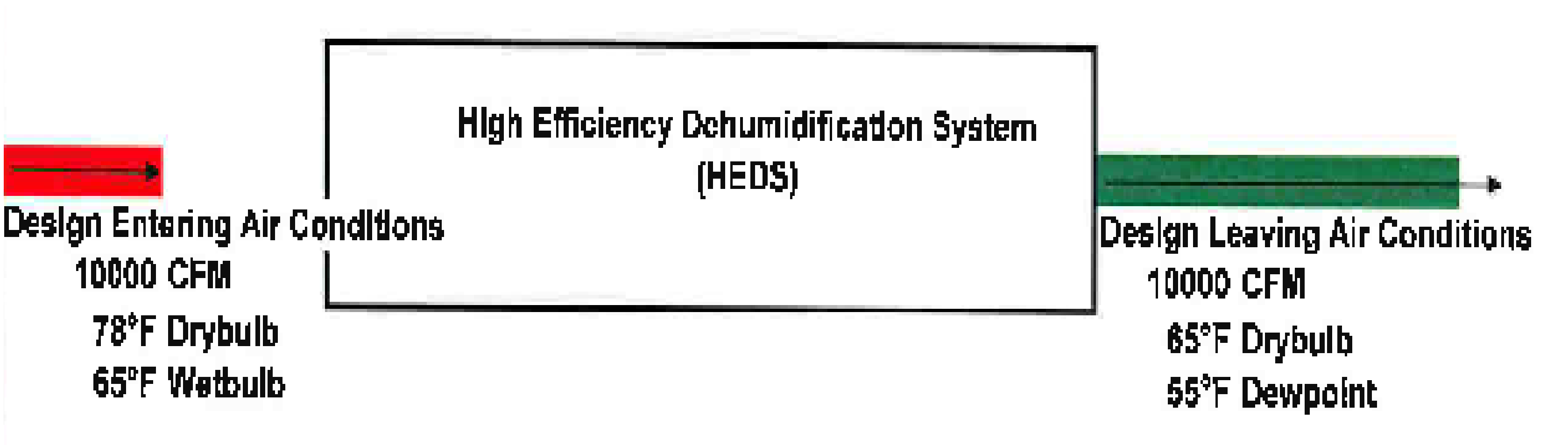
High Efficiency Dehumidification System



Supply air temperature is 10°F above the dewpoint temperature.

For DOAS-HEDS, the supply air temperature can be 20°F above the dewpoint temperature.

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Variable Volume System Performance Comparison

"Normal HVAC" vs. "HEDS"

	Entering Conditions		Leaving Conditions	
% Design CFM	DB	WB	DB	Dewpoint
100%	78	65	65.1	55
75%	77	64.5	65.7	55
50%	76	64	67.2	55
25%	75	63	68.1	55

% Design CFM	Normal AHU System Chiller Plant Load + Reheat Energy (BTUH)	High Efficiency Dehumidification System (HEDS) Chiller Plant Load + Reheat Energy (BTUH)	HEDS % Energy Savings
100%	460013	219240	52%
75%	340795	148330	56%
50%	227500	81250	64%
25%	108160	30400	72%

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DOAS Performance Comparison

"Normal DOAS" vs. "HEDS-DOAS"

	Entering Conditions		Leaving Conditions	
CHWS Temperature at Unit	DB	WB	DB	Dewpoint
45°F	98	82	65	45

CHWS Temperature at Unit	Normal DOAS System Chiller Plant Load + DX Subcooling + Reheat Energy (BTUH)	High Efficiency Dehumidification System (HEDS-DOAS) Chiller Plant Load + DX Subcooling + Reheat Energy (BTUH)	HEDS % Energy Savings
45°F	1561813	1126150	28%

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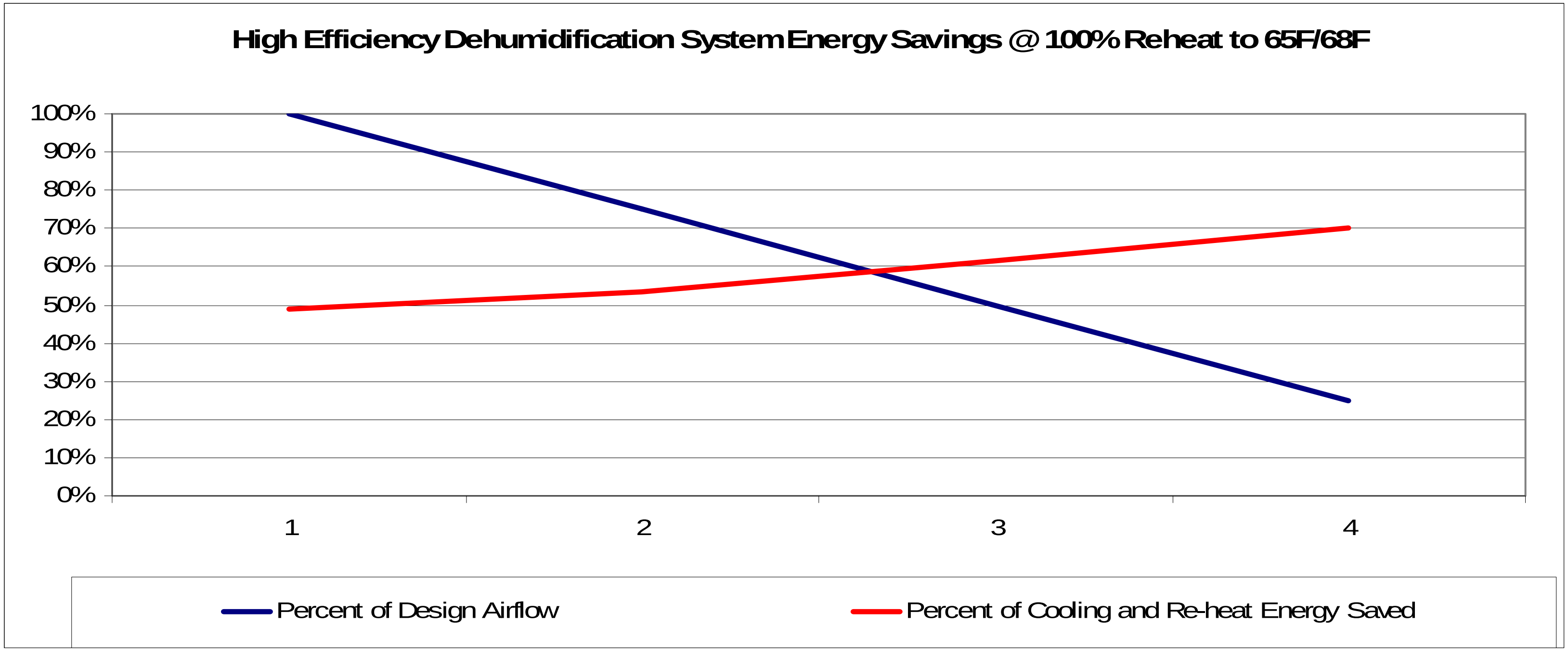
High Efficiency Dehumidification System

Energy Savings vs. Design Airflow @ 100% reheat to 65F/68F – Typical for DOAS, Barracks, Labs, manufacturing, underfloor air distribution systems, etc.



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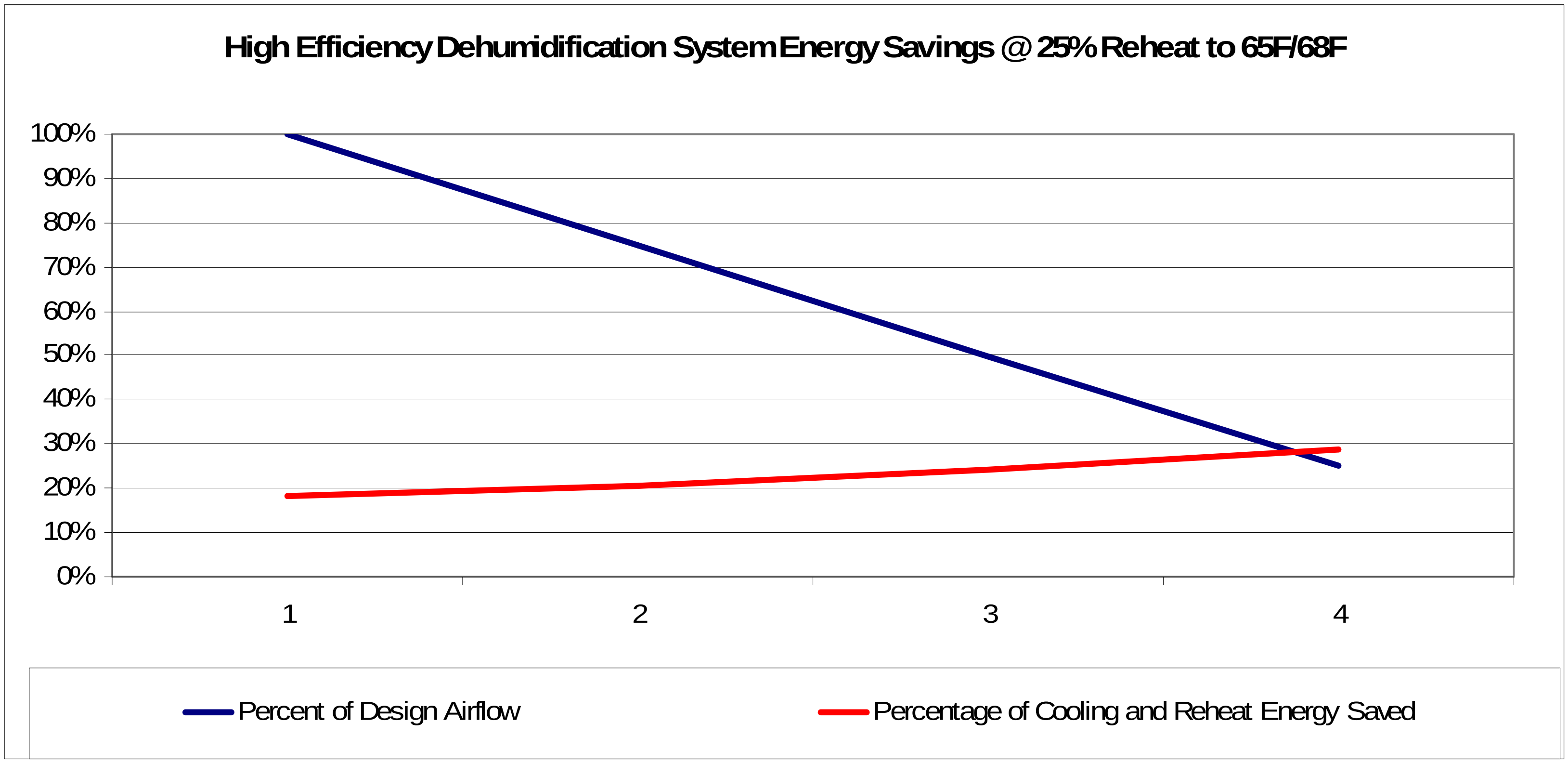
High Efficiency Dehumidification System

Energy Savings vs. Design Airflow @ 25% reheat to 65F/68F – Typical for facilities that need minimal reheat.



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Variable Refrigerant Flow

- For systems that would normally be cooled and heated with DX type systems, variable refrigerant flow (VRF) may be a viable option to consider.
- With proper equipment selections, heating and cooling can occur on the same system, using “waste” heat for the heating source.

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Variable Speed Everything (VSE) Design Strategies

- **We've been designing VSE chiller plants almost exclusively for the past decade, including Absorption based chiller plant systems.**
- **Our designs won six ASHRAE Awards** in 2002. (the last time we applied for any awards)
- In a nutshell, everything in the chiller plant system gets a VSD on it, and we use the Load Based Optimization System **(LOBOS)** to minimize energy waste, and improve system controllability and comfort.

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Pick the Right Optimization Strategy!

- **Central Plant Optimization** routines must look at the site loads – without knowing AHU load information, you cannot maximize energy savings, or respond to load changes properly.
- **AHU Optimization routines are required to minimize energy use and promote occupant comfort! Don't ignore 40% to 50% of the energy consumption of the cooling system!**
- **The System Must be Operator Friendly** – If the person that designed it cannot explain it in an understandable manner to the people that operate the facility, it is too complex.

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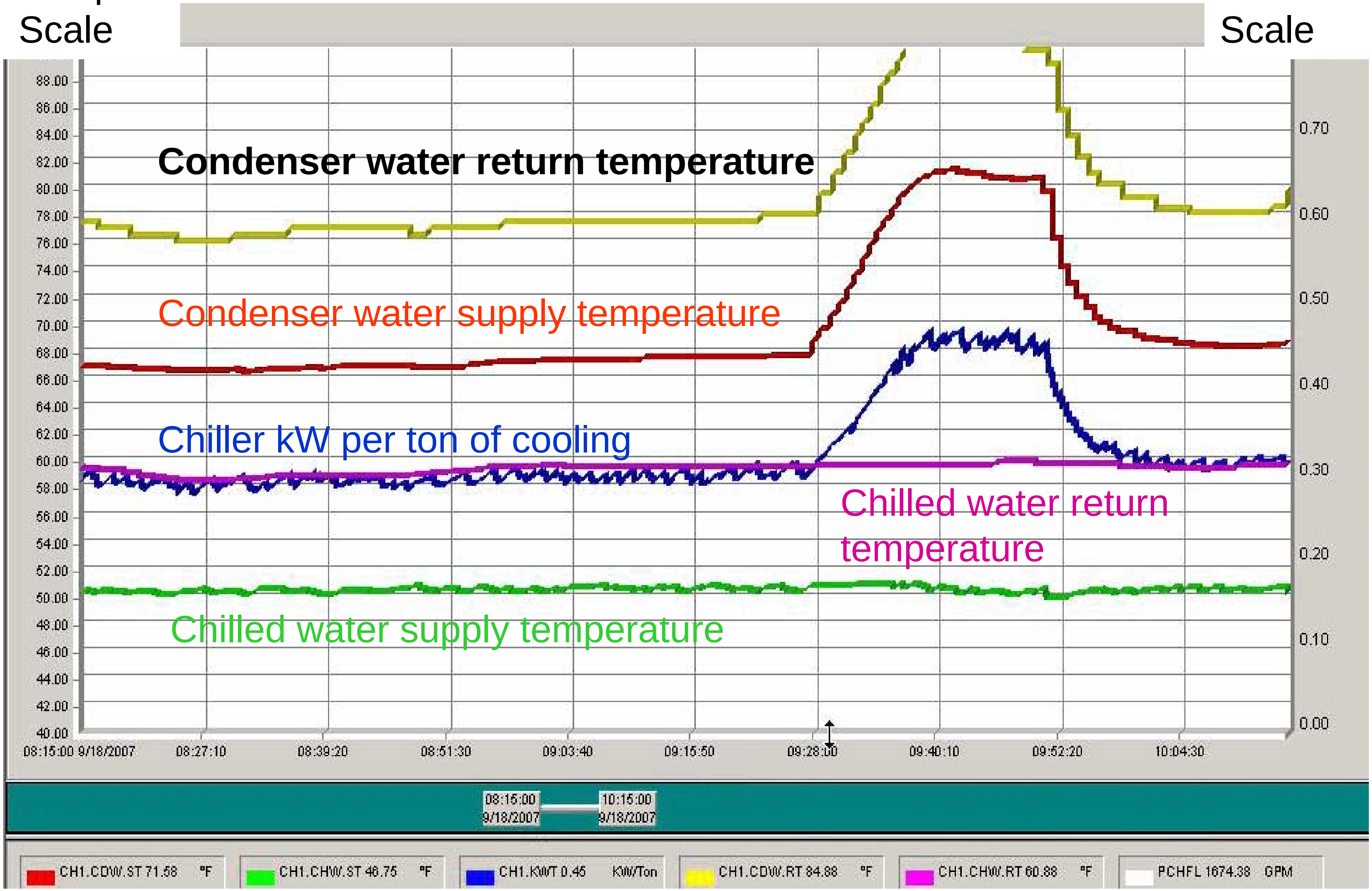
Temp Scale

kW/ton Scale



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Discussion of Previous Page

- This test was run to see what effects changing the condenser water temperature setpoint would have on VFD chiller system efficiency.
- **The red line is the condenser water temperature with a system using the Load Based Optimization System (LOBOS) control system, which determines the most efficient operating points for the HVAC system based on the actual cooling loads of the facility.**
- **The blue line is the chiller efficiency in kW per ton**.
- **The green line is the chilled water supply temperature.**

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Discussion, continued

- The first test, shown in the preceding page, shows that the chiller energy efficiency is running at approximately **0.33 kW per ton**, prior to the start of the test.
- This is excellent efficiency, as most chillers installed today operate between 0.60 and 1.5 kW per ton of cooling.
- With LOBOS, the condenser water temperature is running at approximately 68°F, while the chilled water temperature is running at approximately 50°F.

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- Discussion, continued
- We manually raised the condenser water setpoint to 80°F from the automatically controlled setpoint of 68°F to determine what effect a “normal” operating strategy would have on chiller system performance.
- Many facilities routinely operate their condenser water systems at between 80°F and 85°F, which are the typical “design” points for chillers when they leave the chiller factory.

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Discussion, continued

- As can be seen, the chiller efficiency was made **dramatically worse**, increasing from 0.33 kW per ton to 0.45 kW per ton, **using 36% more energy** to deliver the same amount of cooling. **The chiller energy increased from 332 kW to 452 kW, an increase of 120 kW**. There was a savings in cooling tower energy of approximately 40 kW, but the net effect on the system was an increase of 80 kW, or approximately 21% more energy to provide the same level of cooling.

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- Discussion, continued

- When the condenser water temperature was released back to **LOBOS** operating controls, the efficiency was returned to the previously obtained levels.

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Discussion, continued

- **This also shows that a facility can be designed with excellent cooling equipment, but if it is operated and controlled in a “normal” manner, the efficiency can suffer in a rather dramatic fashion compared to the potential efficiency.**

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Increasing is not equal to Decreasing

- **Normal PID loop controls treat increasing and decreasing loads the same** – if there is an offset from setpoint, the same magnitude and direction response is triggered, whether the load is increasing or decreasing.
- **The Load Based Optimization System (LOBOS) uses incremental control strategies that are configured to respond to increasing and decreasing loads differently.**
- These loops allow more aggressive response while minimizing overshoot, and promoting stable operation.

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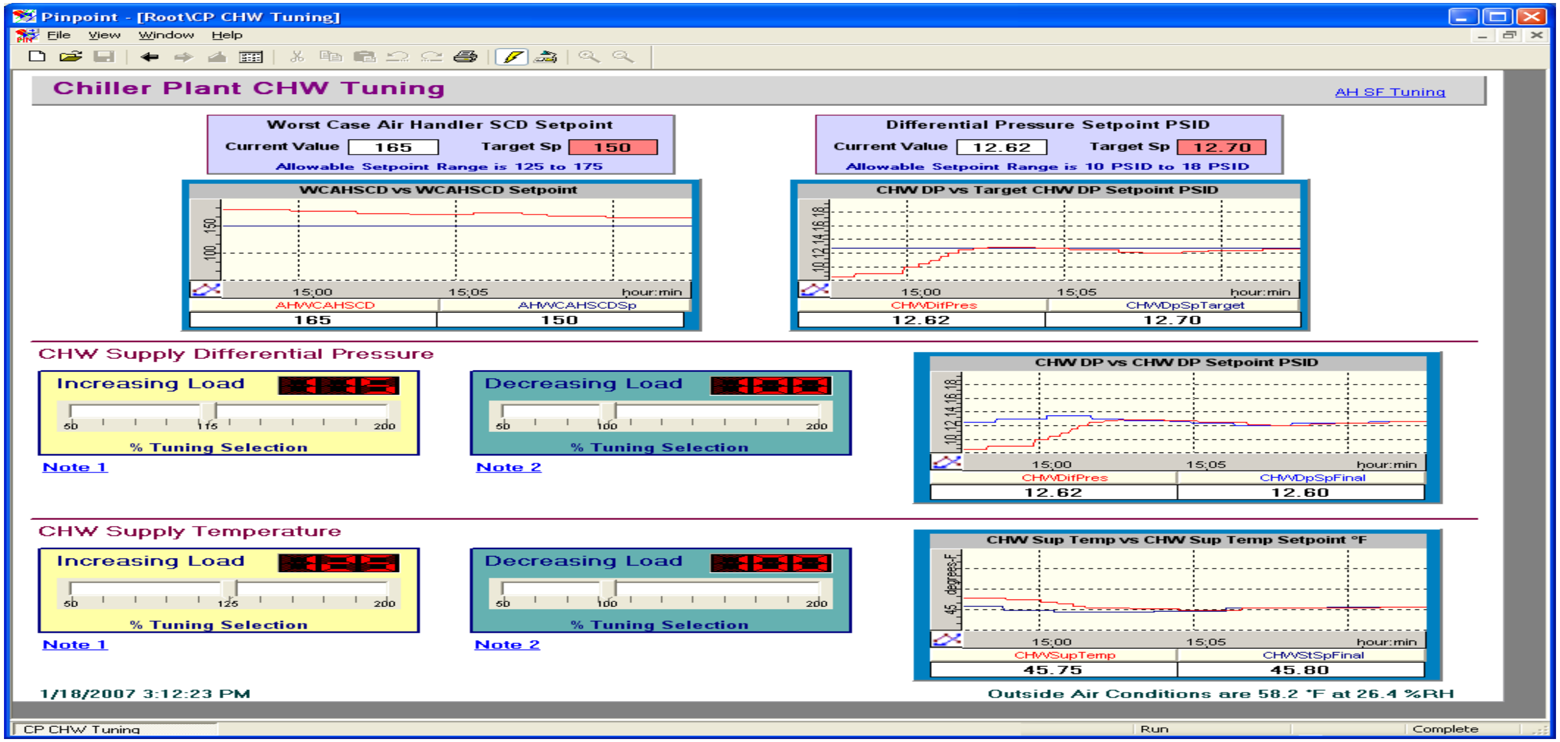
Chiller Plant Tuning Graphic - Simple – Four adjustments - how fast the DP and CHWS temp are adjusted when the loads are increasing and decreasing.



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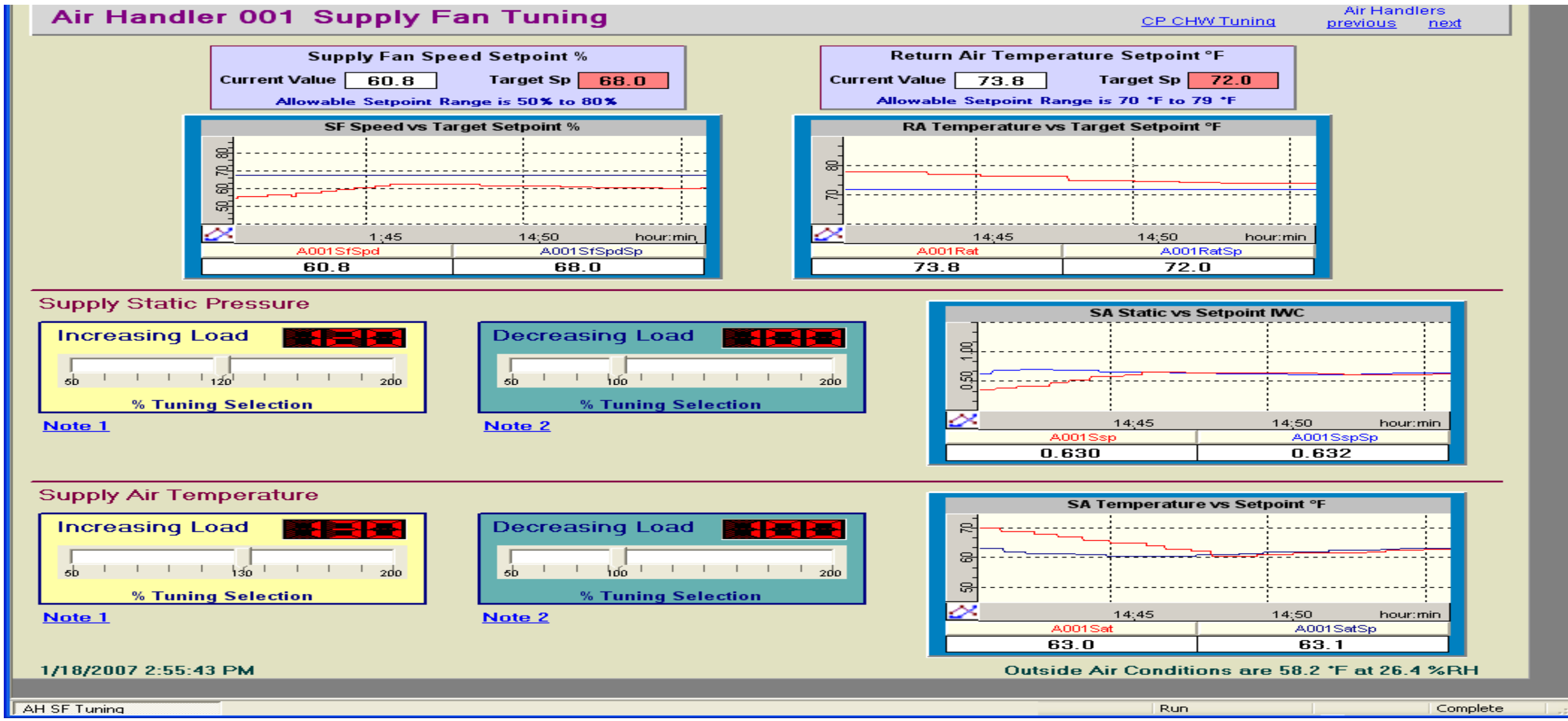
AHU Tuning Graphic - Simple – Six adjustments – Setpoints for fan speed and return air temperature setpoint, and how fast the static pressure and supply air temperature setpoints are adjusted when the loads are increasing and decreasing.



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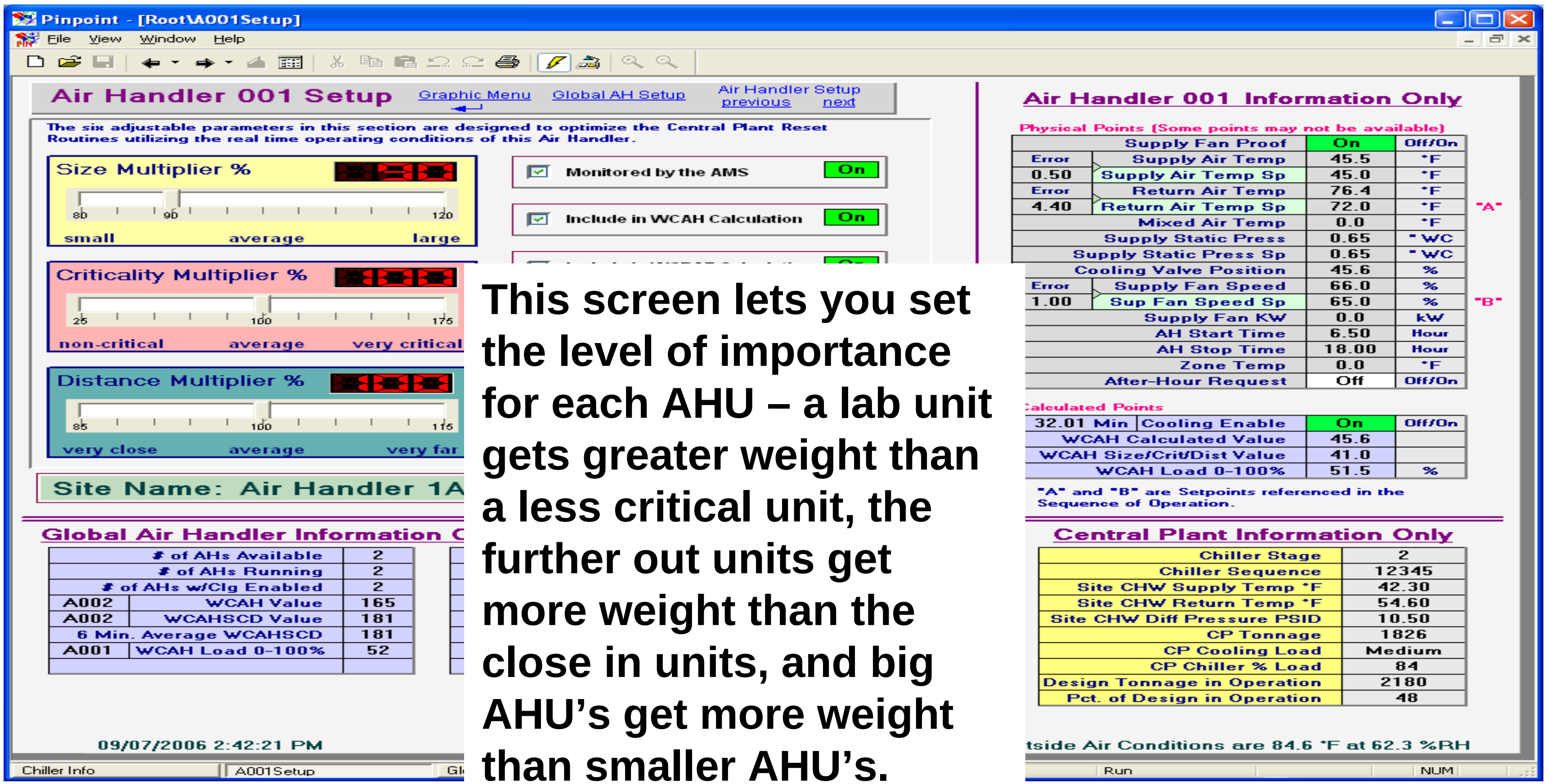
AHU Setup Screen



This screen lets you set the level of importance for each AHU – a lab unit gets greater weight than a less critical unit, the further out units get more weight than the close in units, and big AHU's get more weight than smaller AHU's.

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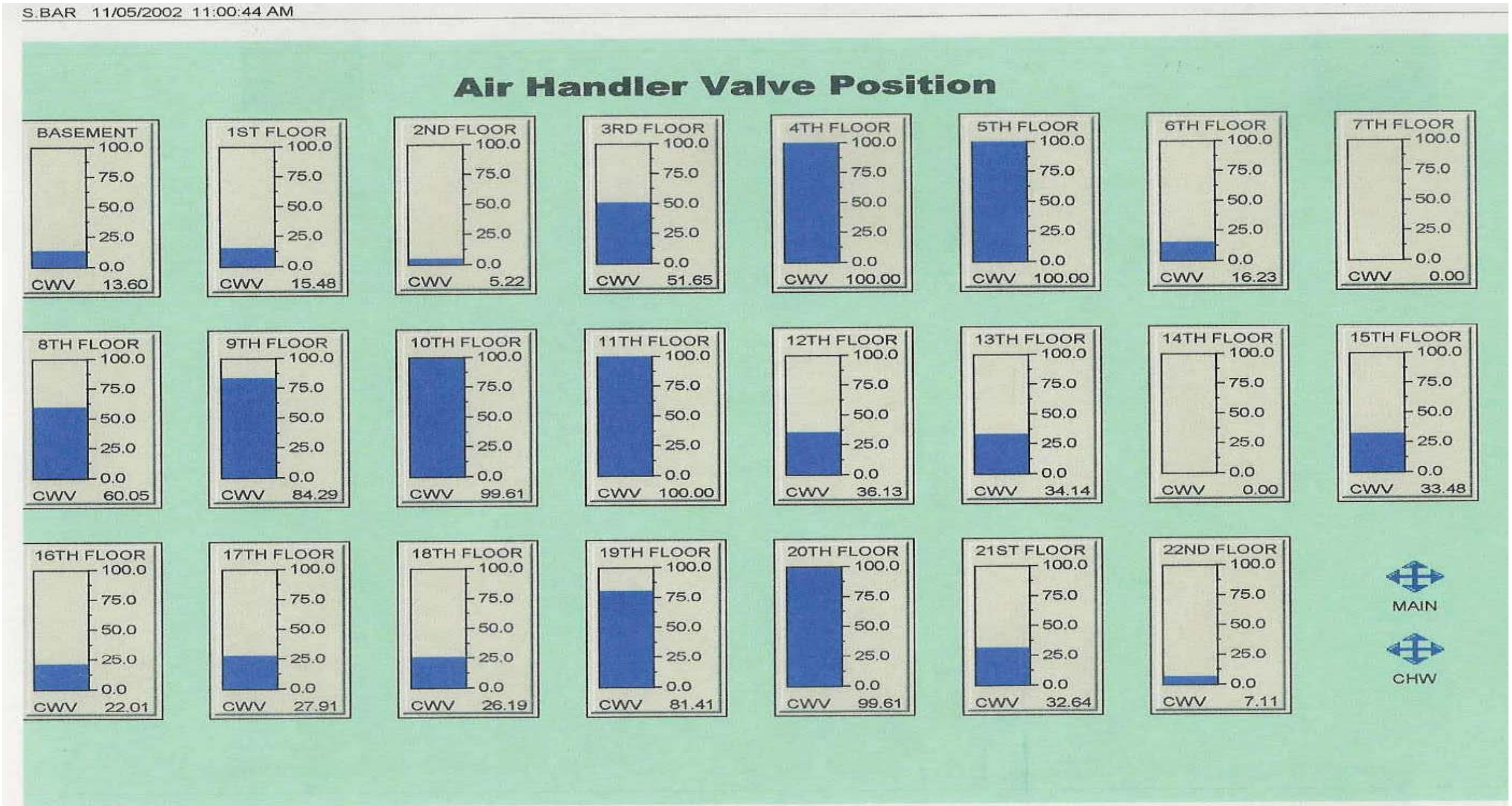
All floors are not loaded alike – the Plant Optimization system must consider all loads!



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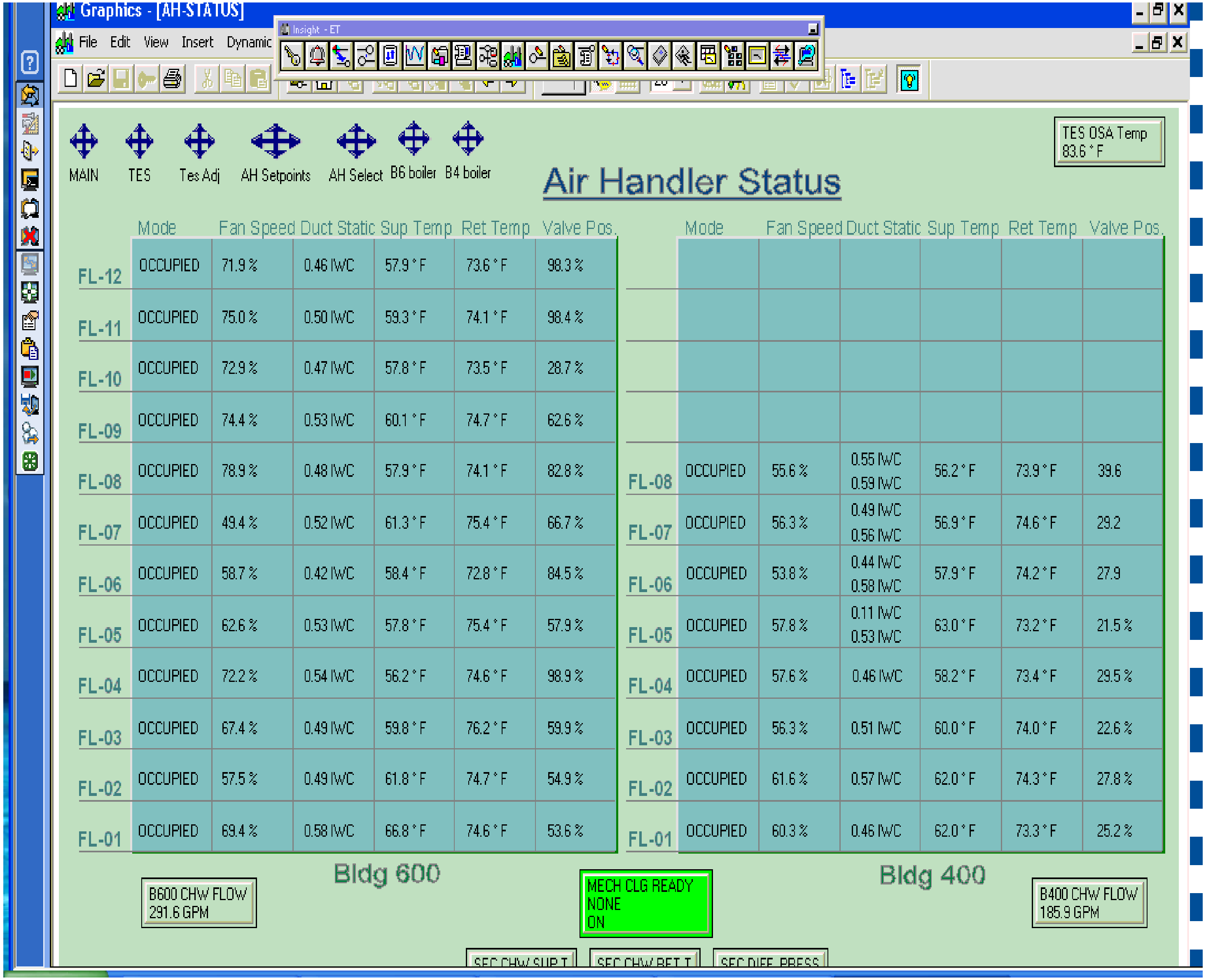

Bldg 600

	Mode	Fan Speed	Duct Static	Sup Temp	Ret Temp	Valve Pos.
FL-12	OCCUPIED	71.9 %	0.46 IWC	57.9 ° F	73.6 ° F	98.3 %
FL-11	OCCUPIED	75.0 %	0.50 IWC	59.3 ° F	74.1 ° F	98.4 %
FL-10	OCCUPIED	72.9 %	0.47 IWC	57.8 ° F	73.5 ° F	28.7 %
FL-09	OCCUPIED	74.4 %	0.53 IWC	60.1 ° F	74.7 ° F	62.6 %
FL-08	OCCUPIED	78.9 %	0.48 IWC	57.9 ° F	74.1 ° F	82.8 %
FL-07	OCCUPIED	49.4 %	0.52 IWC	61.3 ° F	75.4 ° F	66.7 %
FL-06	OCCUPIED	58.7 %	0.42 IWC	58.4 ° F	72.8 ° F	84.5 %
FL-05	OCCUPIED	62.6 %	0.53 IWC	57.8 ° F	75.4 ° F	57.9 %
FL-04	OCCUPIED	72.2 %	0.54 IWC	56.2 ° F	74.6 ° F	98.9 %
FL-03	OCCUPIED	67.4 %	0.49 IWC	59.8 ° F	76.2 ° F	59.9 %
FL-02	OCCUPIED	57.5 %	0.49 IWC	61.8 ° F	74.7 ° F	54.9 %
FL-01	OCCUPIED	69.4 %	0.58 IWC	66.8 ° F	74.6 ° F	53.6 %

Bldg 400

	Mode	Fan Speed	Duct Static	Sup Temp	Ret Temp	Valve Pos.
FL-08	OCCUPIED	55.6 %	0.55 IWC 0.59 IWC	56.2 ° F	73.9 ° F	39.6
FL-07	OCCUPIED	56.3 %	0.49 IWC 0.56 IWC	56.9 ° F	74.6 ° F	29.2
FL-06	OCCUPIED	53.8 %	0.44 IWC 0.58 IWC	57.9 ° F	74.2 ° F	27.9
FL-05	OCCUPIED	57.8 %	0.11 IWC 0.53 IWC	63.0 ° F	73.2 ° F	21.5 %
FL-04	OCCUPIED	57.6 %	0.46 IWC	58.2 ° F	73.4 ° F	29.5 %
FL-03	OCCUPIED	56.3 %	0.51 IWC	60.0 ° F	74.0 ° F	22.6 %
FL-02	OCCUPIED	61.6 %	0.57 IWC	62.0 ° F	74.3 ° F	27.8 %
FL-01	OCCUPIED	60.3 %	0.46 IWC	62.0 ° F	73.3 ° F	25.2 %

LOBOS AHU Resets at work.

OSA Temp is 83.6°F at 3:07 PM.

Average AHU Static pressure is around 0.50”.

Average Supply air temperature is 59.6°F.

Old setpoints were 1.5” to 2” and 48°F to 55°F.

Average fan speed = 61%.

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Subcooling the air due to fan heat added downstream of the coil occurs with this design

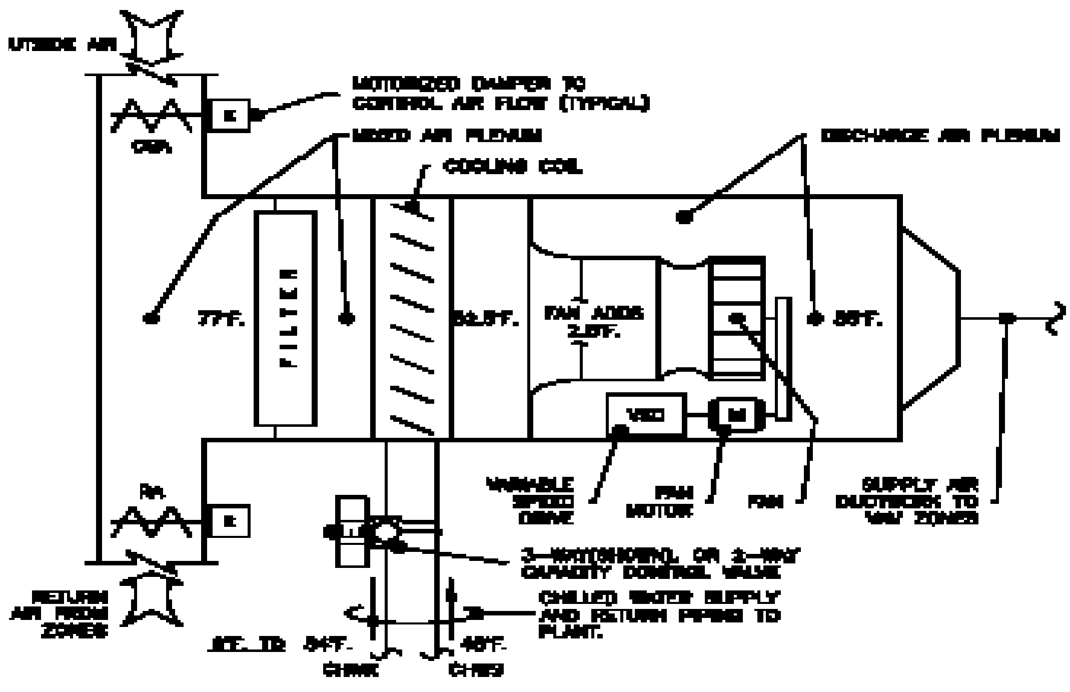
Typical AHU Design

High air and water pressure drops

Typical coil size 4 to 6 row, 550 FPM face velocity

Fixed, or poorly reset, setpoints for static and temperature. 1.5” to 2” and 55°F are typical floor by floor setpoints.

High pressure drop 2-way or 3-way valves



"NORMAL" DESIGN DRAW-THROUGH AIR-HANDLING UNIT (AHU)

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Optimized AHU Design

No subcooling of air due to fan heat added upstream of the coil

CHW TD increased by over 165% (24° vs. 9°)

Desired Coil Size 8 row, 300 to 350 FPM face velocity

Low air and water DP’s

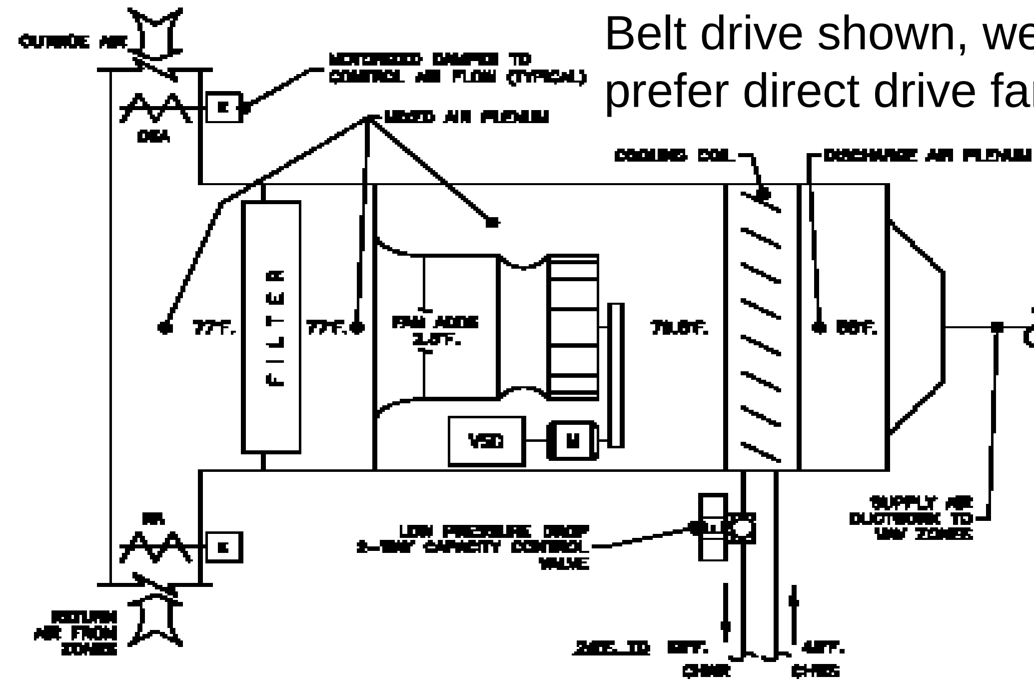
Static and temperature setpoints continually reset based on the loads (LOBOS)

Low pressure drop 2-way valves

Allows chillers in series and substantial pumping and chiller energy savings

COMPARISON CATEGORY	"NORMAL" AHU DESIGN	R.O.I. AHU DESIGN
CHILLED WATER SUPPLY (CHWS) TEMPERATURE / CHILLED WATER RETURN (CHWR) TEMPERATURE / TEMPERATURE SPLIT	45°F. / 54°F. / 9°F.	45°F. / 69.2°F. / 24.2°F.
CHILLED WATER FLOW SAVINGS	—	63%
CHILLED WATER DIFFERENTIAL PRESSURE (DP) SAVINGS	—	21%
AIRSIDE DIFFERENTIAL PRESSURE (DP) SAVINGS	—	49%
PUMP ENERGY SAVINGS	—	70%
(SERIES CHILLERS) CHILLER ENERGY SAVINGS	—	25%
PEAK COOLING LOAD SAVINGS	—	6%

Belt drive shown, we prefer direct drive fans



R.O.I. DESIGN BLOW–THROUGH AIR–HANDLING UNIT (AHU)

SCALE: NONE

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Large Temperature Differential CHW Designs

- **We have many chilled water TES systems that operate with 26°F to 32°F temperature differentials during the summer.**
- Chilled water based TES system storage capacity is nearly directly related to the chilled water temperature differential.
- **A 30 degree TD system will have a tank size 50% smaller than a 15 degree TD system – you can spend some money on coils, and save huge $ on tanks.**

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Large Temperature Differential CHW Designs (cont)

- Large Temperature Differential (LTD) system designs can also save significantly in infrastructure costs.
  - **You can move twice as many tons through a piping system at a 30°F TD than you can with a 15°F TD.**
  - You may not need to add pipes and pumps to meet new loads
  - **It might be less expensive to double your system temperature differential than it is to build new piping and pumping infrastructure, and it is definitely more energy efficient to do this.**

- **A 30°F TD system will cut chilled water pump energy by over 70% when compared with a 15°F system.**

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Strategy to Increase Chilled Water System Temperature differential

- Many facilities are plagued by low chilled water system temperature differentials.
- **This is commonly referred to as the “Low Delta T Syndrome”**
- **This hurts energy performance, whether or not the system has TES installed.**
- **If the system is equipped with properly selected two-way throttling type cooling coil control valves, and the chilled water supply temperature and differential pressure are properly controlled, the most likely culprit for low system temperature differentials is undersized or “worn-out” cooling coils.**

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Increase CHW System TD's

to reduce energy consumption and reduce TES costs

- When we started working as efficiency Consultants for the **University of Southern California** in the early 1990's, the peak day chilled water system temperature differential in the summer was in the **8°F to 9°F** range.
- Over a 16 year period we have been able to work with the Campus to raise the peak day TD to between **24°F and 26°F.**
- **This is a 300% increase in the campus CHW TD!**
- This has been accomplished by replacing cooling coils and control valves, and properly controlling the loads.
- This work cut the required TES tank size from 8,500,000 gallons down to 3,000,000 gallons, **saving approximately $6,000,000 and making TES cost effective.**

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Draw thru vs. Blow thru Cooling Coils

- **Can cooling coil placement (upstream of fan vs. downstream of fan) affect chilled water system temperature differential?**

- **Can cooling coil placement affect total cooling loads?**

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Importance of Coil Selection and Fan Orientation

- 8 row 12 fin per inch 300 to 350 FPM cooling coils are desired for peak system efficiency.
- Blow through coil design to increase CHW system TD and reduce peak loads due to reduction in over-dehumidification.
- Blow thru coils can increase CHW system TD by 34% to 65%+ and decrease peak day cooling loads by 5% or more.

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Cooling Coil Performance Comparisons

- The following pages compare blow thru and draw thru cooling coil selections.
- “Normal” design coils are compared to coils that reduce air and water pressure drop, increase chilled water system temperature differentials and reduce AHU, chiller and pumping system energy consumption.

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Draw Thru 6 R 550 FPM Coil vs.

Blow Thru 8 R 350 FPM Coil

	Typical Design	**Near-Optimal Design**
	Draw Thru 6 Row 550 FPM	Blow Thru 8 Row 350 FPM
Mixed Air Temperature DB	77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F
Coil Entering WB	64.8°F	66°F
Coil Leaving DB	52.5°F	55°F
Coil Leaving WB	51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55
Chilled Water Supply Temperature (°F)	45	45
Chilled Water Return Temperature (°F)	54	69.2
Chilled Water Temperature Differential (°F)	9	24.2
CHW TD % Change		**169%**
Cooling Coil Water DP (ft H2O)	10.1	8
CC DP % Change		**-21%**
Cooling Coil Air DP (InWC)	1.26	0.64
CC DP % Change		**-49%**
Coil GPM Required	70	24.4
Coil GPM % Change		**-65%**
Total BTUH	315000	298000
BTUH % change		**-6%**

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6R 550 FPM Draw thru vs Blow thru and 8R 550 FPM Draw thru vs Blow thru

	Draw Thru 6 Row 550 FPM	**Blow Thru 6 Row 550 FPM**		Draw Thru 8 Row 550 FPM	**Blow Thru 8 Row 550 FPM**
Mixed Air Temperature DB	77°F	77°F		77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F		64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F		77°F	79.5°F
Coil Entering WB	64.8°F	66°F		64.8°F	66°F
Coil Leaving DB	52.5°F	55°F		52.5°F	55°F
Coil Leaving WB	51.5°F	54°F		51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55		55	55
Chilled Water Supply Temperature (°F)	45	45		45	45
Chilled Water Return Temperature (°F)	54	60.2		58.6	64.7
Chilled Water Temperature Differential (°F)	9	15.2		13.6	19.7
CHW TD % Change		**69%**			**45%**
Cooling Coil Water DP (ft H2O)	10.1	9.9		9.1	8.7
CC DP % Change		**-2%**			**-4%**
Cooling Coil Air DP (InWC)	1.26	1.14		1.58	1.43
CC DP % Change		**-10%**			**-9%**
Coil GPM Required	70	39		46.4	30
Coil GPM % Change		**-44%**			**-35%**
Total Peak Load BTUH	315000	296000		315000	296000
BTUH % change		**-6%**			**-6%**

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6R 350 FPM Draw thru vs Blow thru and 8R 350 FPM Draw thru vs Blow thru

	Draw Thru 6 Row 350 FPM	**Blow Thru 6 Row 360 FPM**		Draw Thru 8 Row 350 FPM	**Blow Thru 8 Row 350 FPM**
Mixed Air Temperature DB	77°F	77°F		77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F		64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F		77°F	79.5°F
Coil Entering WB	64.8°F	66°F		64.8°F	66°F
Coil Leaving DB	52.5°F	55°F		52.5°F	55°F
Coil Leaving WB	51.5°F	54°F		51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55		55	55
Chilled Water Supply Temperature (°F)	45	45		45	45
Chilled Water Return Temperature (°F)	60	65.5		63.1	69.2
Chilled Water Temperature Differential (°F)	15	20.5		18.1	24.2
CHW TD % Change		**37%**			**34%**
Cooling Coil Water DP (ft H2O)	10.5	8.5		8.2	8
CC DP % Change		**-19%**			**-2%**
Cooling Coil Air DP (InWC)	0.56	0.51		0.71	0.64
CC DP % Change		**-9%**			**-10%**
Coil GPM Required	42	28.9		34.9	24.4
Coil GPM % Change		**-31%**			**-30%**
Total Peak Load BTUH	315000	296000		3´5000	295000
BTUH % change		**-6%**			**-6%**

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Draw Thru 6 R 550 FPM vs. Blow Thru 8 R 350 FPM for 100% OSA system

100% OSA System	45°F CHWS Temp	45°F CHWS Temp	42°F CHWS Temp	42°F CHWS Temp
	Draw Thru 6 Row 550 FPM **(Cannot meet required temps)**	Blow Thru 8 Row 350 FPM	Draw Thru 6 Row 550 FPM	Blow Thru 8 Row 350 FPM
Mixed Air Temperature DB	92°F	92°F	92°F	92°F
Mixed Air Temperature WB	78°F	78°F	78°F	78°F
Coil Entering DB	92°F	94.5°F	92°F	94.5°F
Coil Entering WB	78°F	79.2°F	78°F	79.2°F
Coil Leaving DB	53.1°F	55.8°F	52.5°F	55°F
Coil Leaving WB	52.1°F	54.6°F	51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55.6	55.8	55	55
Chilled Water Supply Temperature (°F)	45	45	42	42
Chilled Water Return Temperature (°F)	49	69	48.5	71
Chilled Water Temperature Differential (°F)	**4**	24	**6.5**	29
CHW TD % Change		**600%**		**346%**
Cooling Coil Water DP (ft H2O)	44	12.6	19.2	13.5
CC DP % Change		**-71%**		**-30%**
Cooling Coil Air DP (InWC)	1.38	0.83	1.38	0.82
CC DP % Change		**-40%**		**-41%**
Coil GPM Required	**362**	59.5	226	50
Coil GPM % Change		**-84%**		**-78%**
Total BTUH	724000	714000	735000	726000
BTUH % change		**-1%**		**-1%**

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TES Optimized Cooling Coil

32 Degree TD – 200% of Typical TES System Designs – Doubles the Capacity of Most Installed TES Systems

	TES Duty 8R 350 FPM
Mixed Air Temperature DB	77°F
Mixed Air Temperature WB	64.8°F
Coil Entering DB	79.5°F
Coil Entering WB	66°F
Coil Leaving DB	55°F
Coil Leaving WB	54°F
Air Handling Unit Supply Air Temperature (°F)	55
Chilled Water Supply Temperature (°F)	40
Chilled Water Return Temperature (°F)	72.1
Chilled Water Temperature Differential (°F)	32.1
CHW TD % Change	
Cooling Coil Water DP (ft H2O)	4.9
CC DP % Change	
Cooling Coil Air DP (InWC)	0.61
CC DP % Change	
Coil GPM Required	18.4
Coil GPM % Change	
Total BTUH	296000
BTUH % change	

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30 Degree + TD’s are Possible!

- **We have many systems that operate with 30°F to 32°F chilled water system temperature differentials.**
- It can be done, and it is not that big of a deal to make it happen.
- Big coils allow lots of leeway for energy conservation, and also allow loads to change without having to spend $$$ to upgrade the HVAC system.

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“Normal” new T24 Chiller Plant at real world conditions.

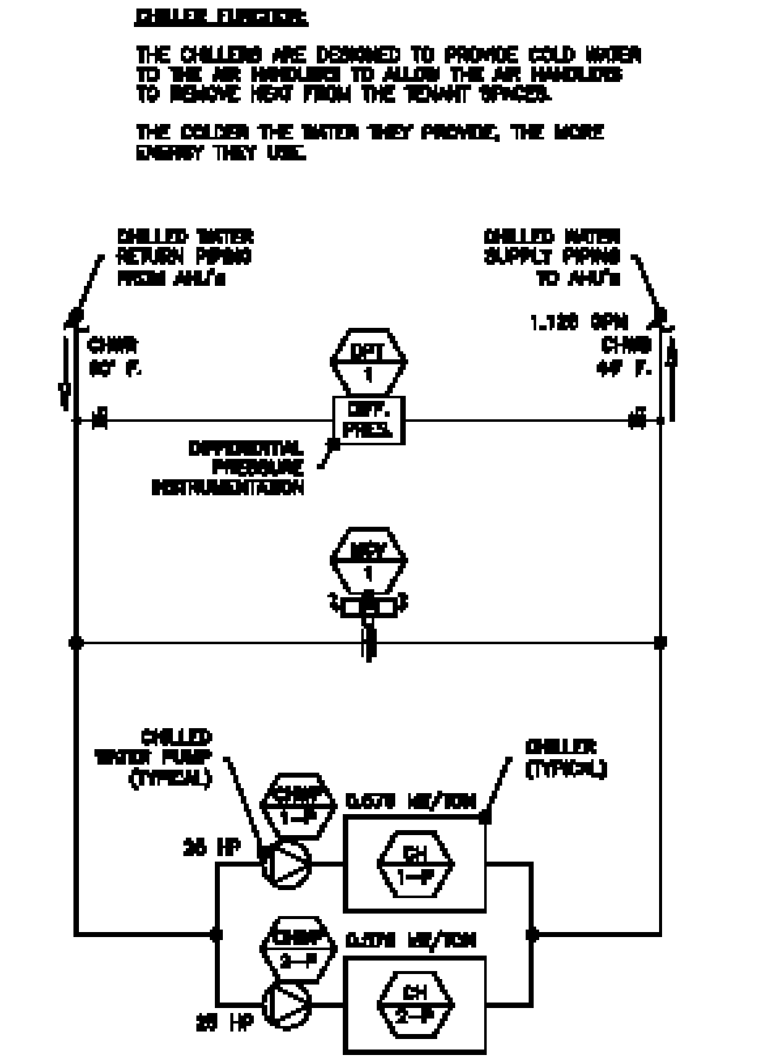
Chillers in parallel, dedicated pumps, 44°F to 60°F chilled water temps.

Combined chiller efficiency = 0.576 kW/ton @ full load.

Chiller + CHW pump energy = 0.629 kW/ton @ full load

Typically, fixed or minimally reset CHW supply temperature and Differential Pressure setpoints.

All piping arranged in a “T” configuration, to maximize turbulence and wasted energy.



CHILLED WATER SYSTEM, CHILLERS IN PARALLEL, TYPICAL DESIGN

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“Easily Affordable” new High Efficiency T24 Chiller Plant at real world conditions.

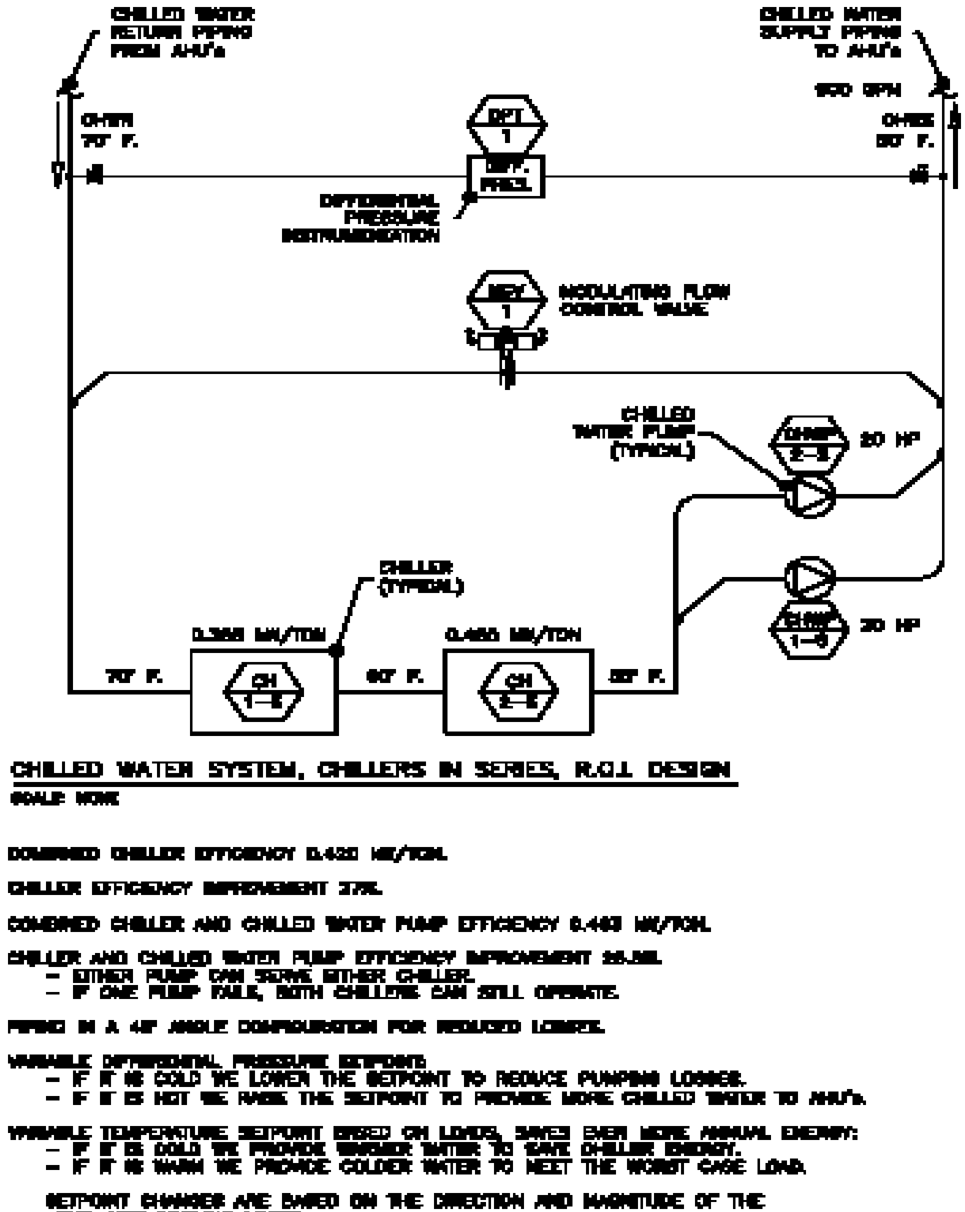
Chillers in series, header’d pumps, 50°F to 70°F chilled water temps.

Combined chiller efficiency = 0.420 kW/ton @ full load. (27% improvement)

Chiller + CHW pump energy = 0.463 kW/ton @ full load (26.5% improvement)

LOBOS reset strategies for CHW supply temperature and Differential Pressure setpoints.

All piping arranged in a 45° angle configuration, to minimize turbulence and wasted energy.



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Typical Southern California Cooling Tower System

72°F Wetbulb temp design

Dedicated, constant speed pumps

Fixed, or manually variable condenser water setpoint

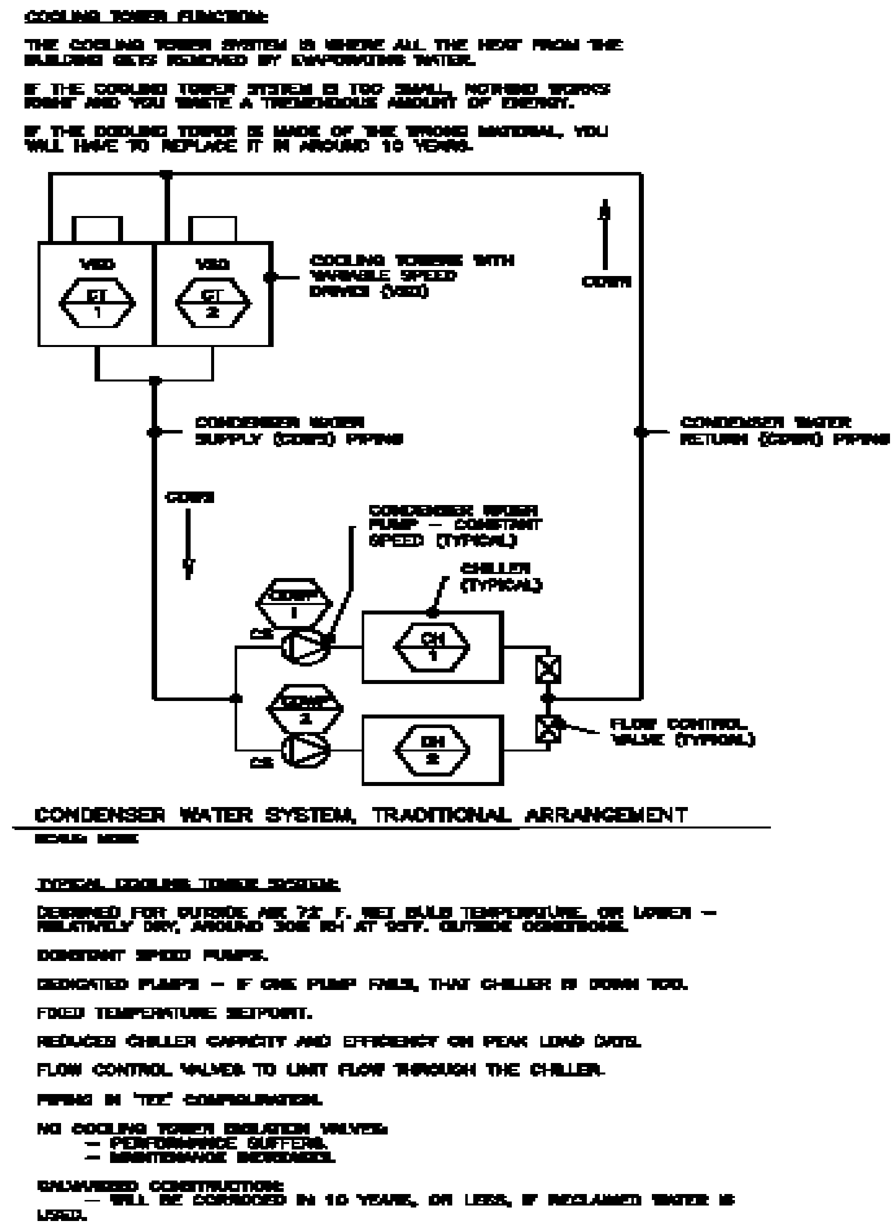
Flow control valves on condenser barrels

Higher condenser water temperatures hurt chiller performance.

All piping arranged in a "T" configuration, to maximize turbulence and wasted energy.

No automated CT cell isolation valves hurts performance and increases maintenance costs.

Galvanized construction will be eaten by reclaimed water in most places (if reclaimed water is required)



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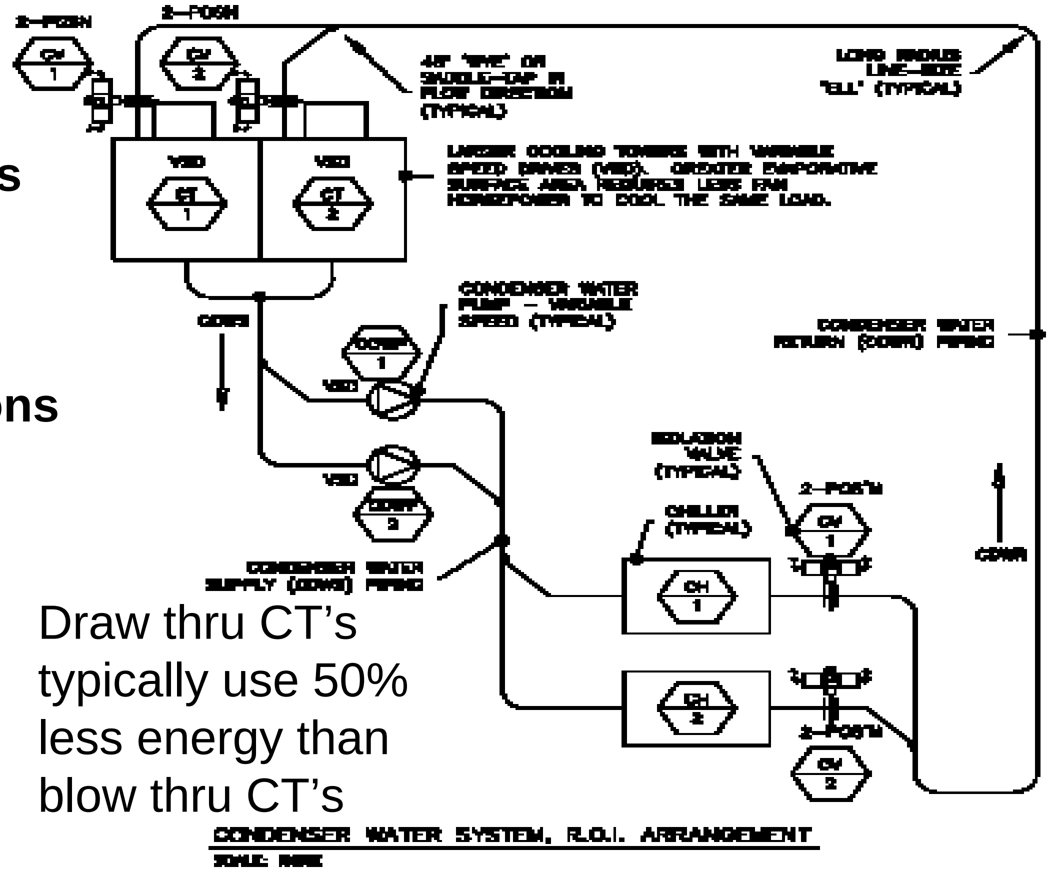
78°F Wetbulb temp design.



Header'd, variable speed pumps

Continually variable condenser water setpoint, based on load, efficiency and ambient conditions

No flow control valves on condenser barrels

Draw thru CT's typically use 50% less energy than blow thru CT's

Lower condenser water temperatures improve chiller performance (within limits).

Larger surface area reduces CT fan energy requirements.

Automated CT cell isolation valves help performance and decrease maintenance costs.

All piping arranged in a 45° configuration, to minimize turbulence and wasted energy.

316 SS construction will provide a 25 to 30 year service life, and is more compatible with reclaimed water in most places (if reclaimed water is required)

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Reset Logic - AHU

- Look at the load at the zone or floor level:
  - **Reset the static pressure setpoint and the supply air temperature setpoint based on the loads, and whether the loads are increasing or decreasing.**
  - **Increasing loads should be handled differently than decreasing loads.**
  - **Use “Thermal Flywheel” logic to reduce overall system energy consumption.**

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Reset Logic - CHW Side

- Look at the load at each Air Handler:
  - **Reset the chilled water system differential pressure setpoint and the chilled water supply temperature setpoint based on the loads, and whether the loads are increasing or decreasing.**
  - **Increasing loads should be handled differently than decreasing loads.**
  - **Use "Thermal Flywheel" logic to reduce overall system energy consumption.**

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Reset Logic - CDW Side

- Look at the load on the chillers:
  - **If the load is increasing, increase the condenser water pump flow to decrease chiller energy.**
  - **If the load is decreasing, decrease the condenser water pump flow to decrease pump energy and increase CT efficiency.**
  - **If the load is low, decrease the condenser water supply temperature to improve chiller performance.**
  - **VFD Chillers must be treated differently than constant speed chillers.**
  - **If the load is high, allow the condenser water temperature to float to respond to loads, efficiency, and ambient conditions to improve system performance.**

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Interim Summary

- **It is not that tough to save a substantial amount of energy and reduce GHG emissions, while at the same time improving comfort conditions.**
- **Proper equipment and system sizing and selection should be based on the lifecycle cost of the facility, and promoting comfort and IAQ.**
- **User friendly HVAC optimization controls promote persistent energy efficiency.**
- **The bottom line will be up to your operating staff, if they can tune the control system to suit the specific needs of the loads being served, the system has a far greater chance of success than if the system cannot be easily tuned.**

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“Normal” duct and VAV box sizing

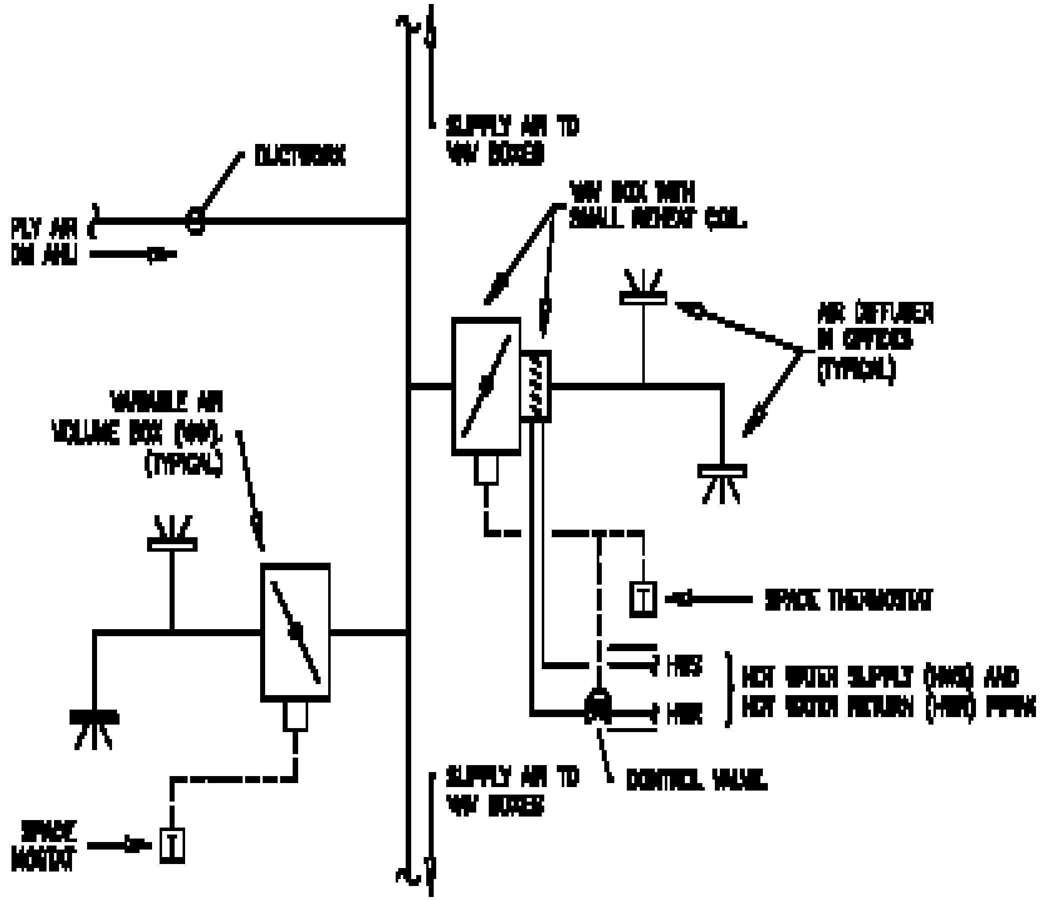
Small reheat coil with high air pressure drop

Box requires 1.5” at inlet

Pneumatic Controls

Reheat coil sized for 180°F/140°F water temps

Boiler efficiency around 80% to 84%



VARIABLE AIR VOLUME (VAV) BOX AND VAV BOX WITH REHEAT.

SCALE: NONE

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Larger duct and VAV box sizing reduces loop pressure drop by 20% to 35%.

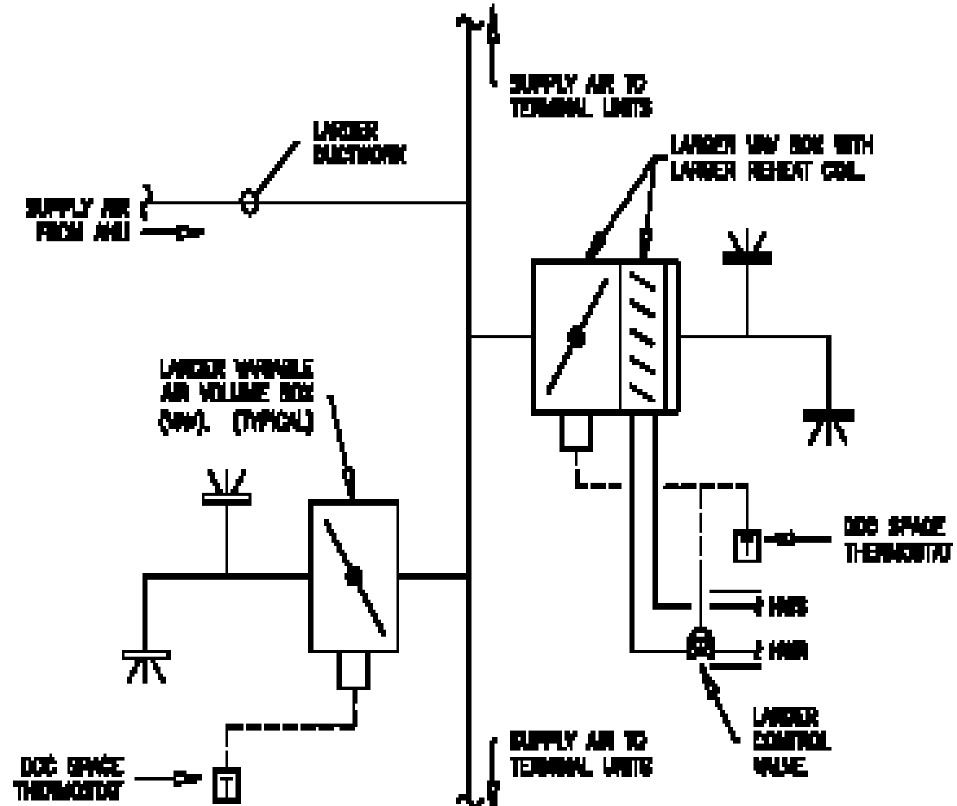
Large face area reheat coil with low air pressure drop

Box requires 1.0” at inlet

DDC Controls with LOBOS feedback to AHU

Reheat coil sized for 120°F/80°F water temps

Boiler efficiency around 90% to 96%



VARIABLE AIR VOLUME (VAV) BOX AND VAV BOX WITH REHEAT.

SCALE: NONE

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Develop and Follow a Strict Design Guideline for the HVAC System

- We have developed and provided design guidelines for many clients to ensure that their facility's energy efficiency improves over time, rather than degrades.
- The design guidelines are incorporated into all new facilities, facility expansions and remodeling/rebuilding projects.
- The guidelines are handed to the Architects and Engineers and they are strictly enforced, typically with lots of objections by the A/E team.
  - **Why do high efficiency, comfort inducing systems, that the Owner is requesting, cause Engineers to bristle?**

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ESAC Case Study

- **900,000 SF facility**
- **Pre-modification on-peak cooling loads – 1,800 tons + partial storage TES system.**
- **AHU cooling coils replaced with 28 degree TD coils**
- **2,400 ton P/S plant upgraded to VFD-everything, and to 2,600 tons, with “Oversized” cooling towers.**
- **Chillers re-piped in series.**
- **TES system re-piped with new tank headers and plant piping – 350% more TES capacity.**
- **New DDC control system installed with complete loadside optimization routines.**

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ESAC Case Study, continued

- We added a 230,000 SF building, and removed an 800 ton constant speed chiller, installing a new 1,420 ton VFD chiller, **piped in series at 0.37 kW per ton at full load.**
- **The 1,420 ton chiller uses less power than the 800 ton chiller it is replacing, while producing 78% more cooling capacity.**
- All 1,125,000 square feet will still be served by the now full storage TES system.

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ESAC Case Study, continued

- The project saved over $1,000,000 by not having to build a new chiller plant to serve the new building.
- The project reduced operating costs by an average of $500,000 per year.

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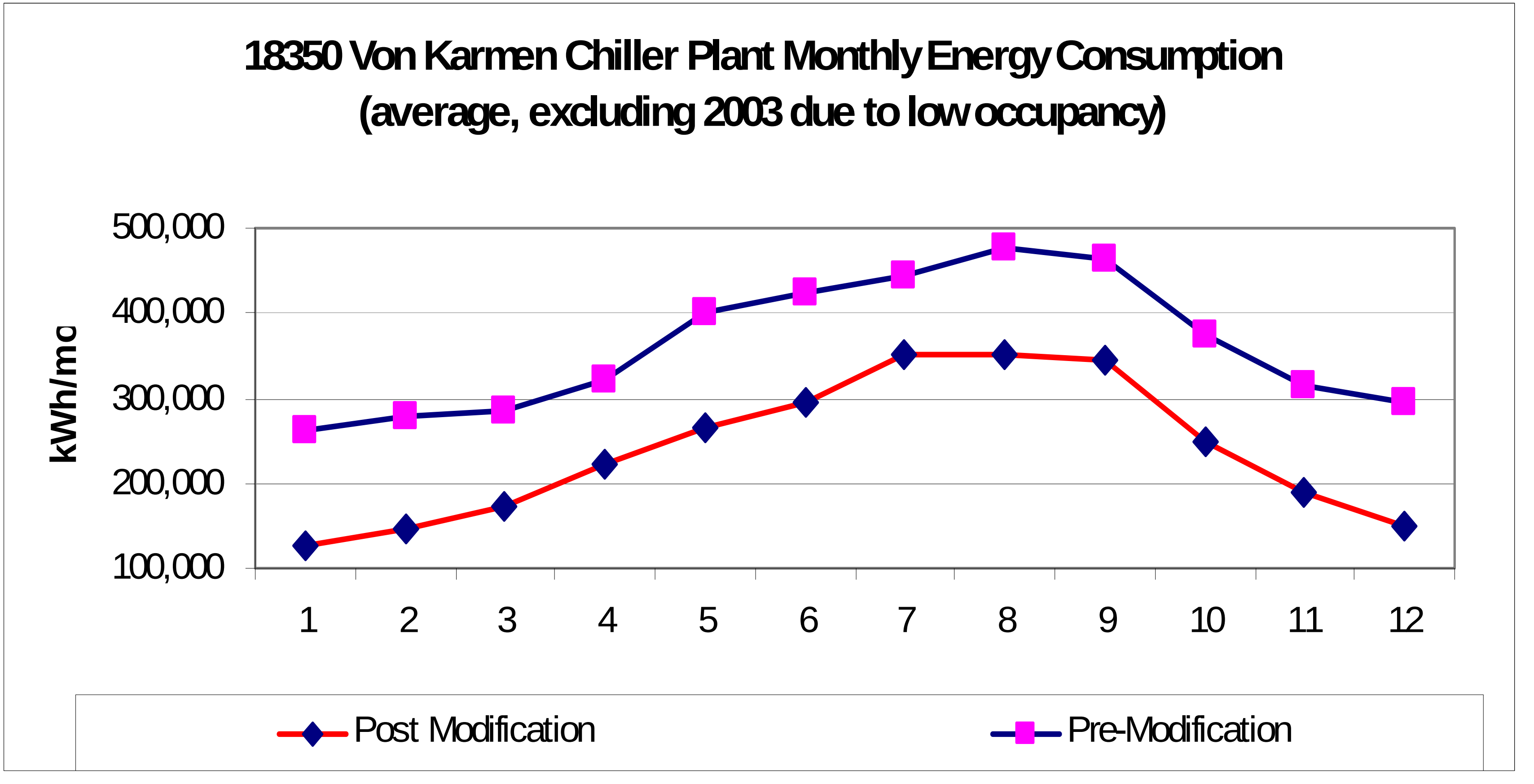
Monthly Average kWh Consumption for Pre and Post TES System Upgrade 35% Annual Savings (From Utility Meter)



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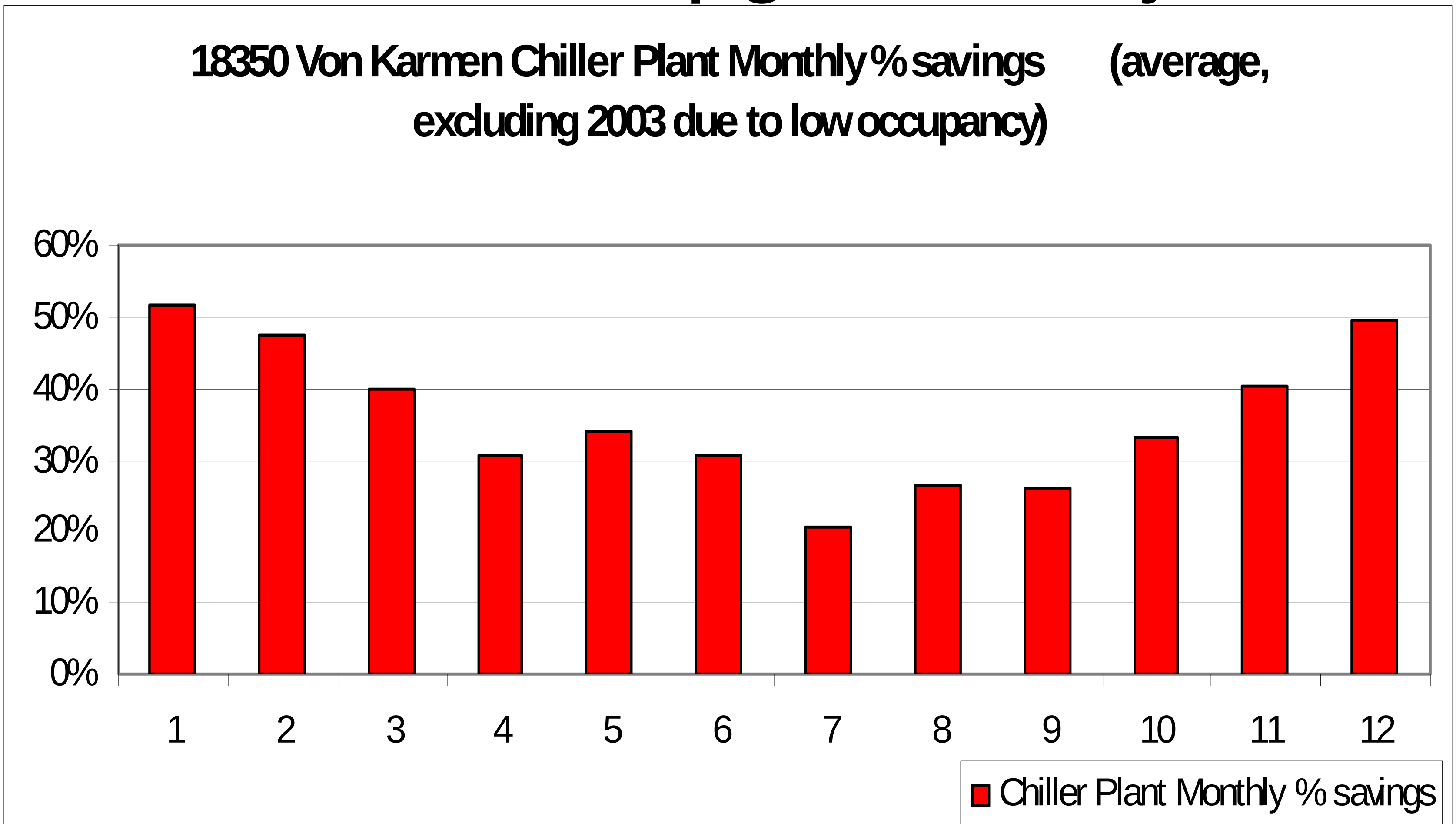
Average Monthly Energy Savings From TES Upgrade Project



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kW per Ton Can Be Misleading!

- **You can have a plant system that seems to show great efficiency, with a low kW per ton, but kW per ton is not the way owners pay for electricity!**
- **If the plant is made to look good by raising the CHW temperature too far, the supply fan energy can escalate off the charts – but the plant still looks great!**

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kW per Ton Can Be Misleading!

- **If the AHU resets are running the SAT too low, there will be added dehumidification loads, which will increase the tonnage on the plant, and make the system kW/ton look better than it is, since the total kW consumed by the HVAC system may be higher than it should be.**

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Extreme Low Temp Air Test

- **The following slide depicts a system that is operating at 96% of the typical peak load (480 tons on a 500 ton chiller).**
- **The entire HVAC system, including chiller, all pumps, cooling tower and supply and return fans, is operating at 0.49 kW per ton.**

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Is 0.49 kW/ton for the Entire HVAC System Good?

- **Most people would think that this is fantastic performance, and if that were the only information available, I would strongly agree.**
- **It is extremely rare to see a chiller system operating at less than 0.49 kW per ton at 96% load, let alone an entire HVAC system.**

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Low Temp Test, Continued

- **The factor that is not shown in the 0.49 kW per ton figure is that the tons were dramatically increased due to the dehumidification load associated with dropping the supply air temperature down to 39°F.**

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Low Temp Test, Continued

- **The overall energy demand and consumption went up, while the kW per ton for the system went from around 0.65 kW per ton down to 0.49 kW per ton.**
- **If you just looked at the kW per ton figure, you would think that you were a star! A 25% improvement! You get a gold medal! Wrong!**

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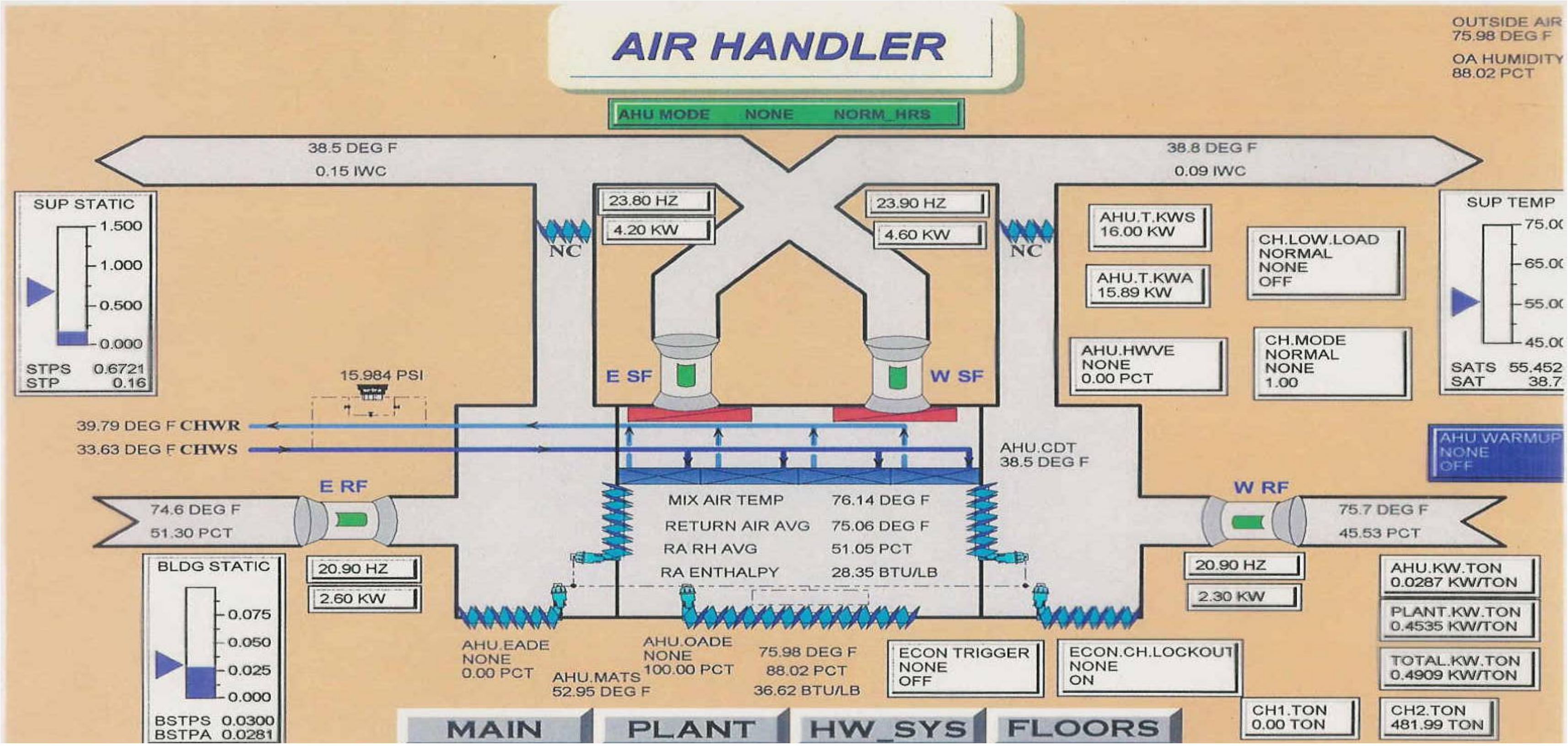
Low Temp Air – Extreme Test - 39°F Air, 0.15 INWC static pressure
0.03 AHU kW/Ton, 0.49 Total HVAC system kW/Ton



While this looks impressive from a kW/ton basis, the dehumidification load to provide 39°F air temperatures increased the tonnage dramatically, so overall energy use went up, not down.

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Don't just look at kW per ton! It can be extremely misleading!

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Put the VFD on the Biggest Chiller!

- **VFD chiller performance is related to refrigerant lift and relative load.**

- **Refrigerant lift is related to chilled and condenser water temperatures and refrigerant approach temperatures.**
- **Refrigerant approach temperatures are related to relative load and heat transfer surface area.**
- **Relative load and heat transfer surface area are related to the size of the chiller.**
  - **The bigger the chiller, the lower the relative load will be and the greater the heat transfer surface area will be.**
- **The lower the relative load and the greater the heat transfer surface area, the closer the approach temperatures can be.**
- **The closer the approach temperatures can be, the lower the lift can be.**
- **The lower the lift, the more efficient the chiller can be (within limits).**

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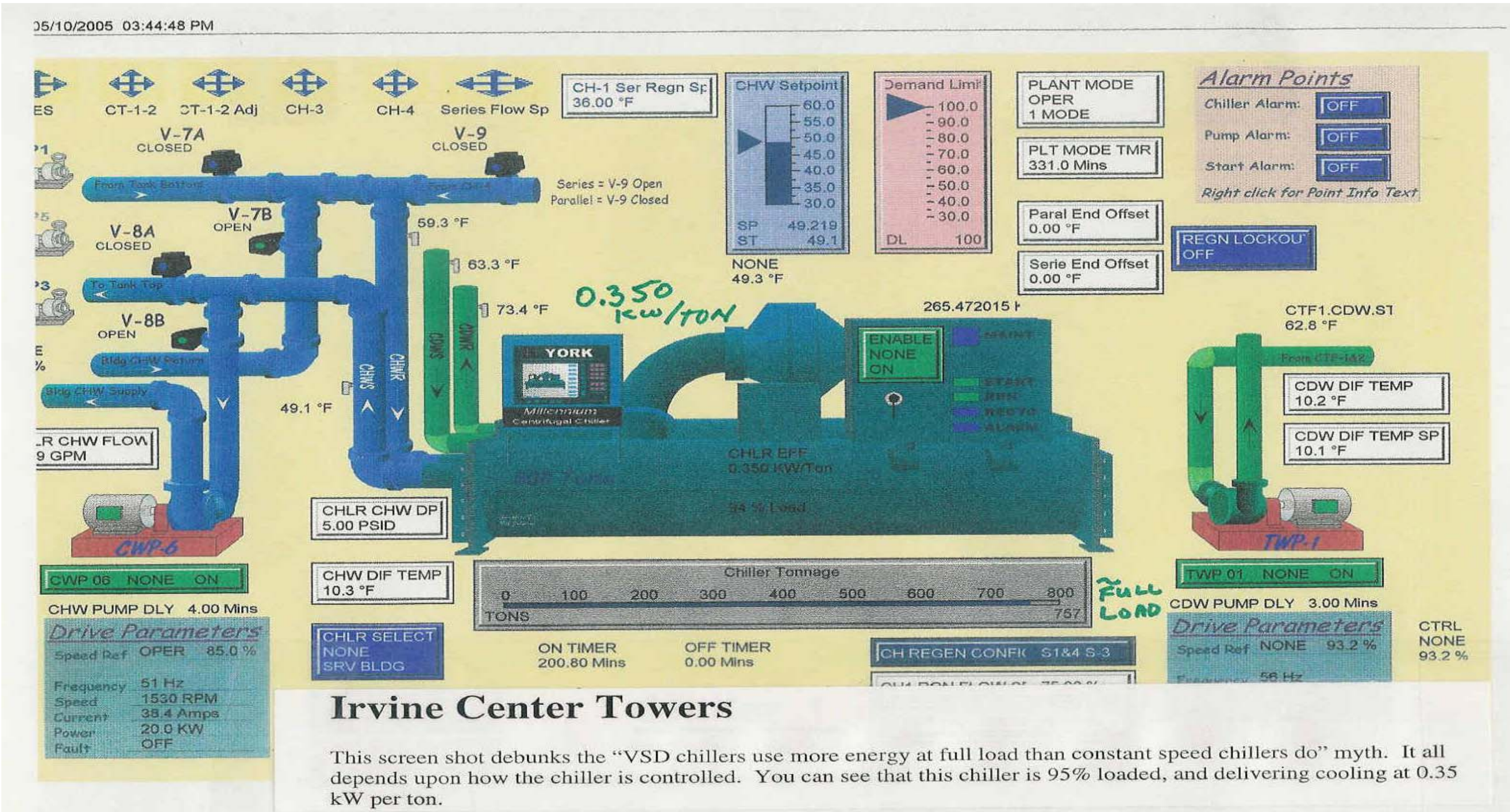
Fully Loaded VFD chiller @ 0.35 kW/Ton – VFD Chiller Efficiency is more related to lift than to load



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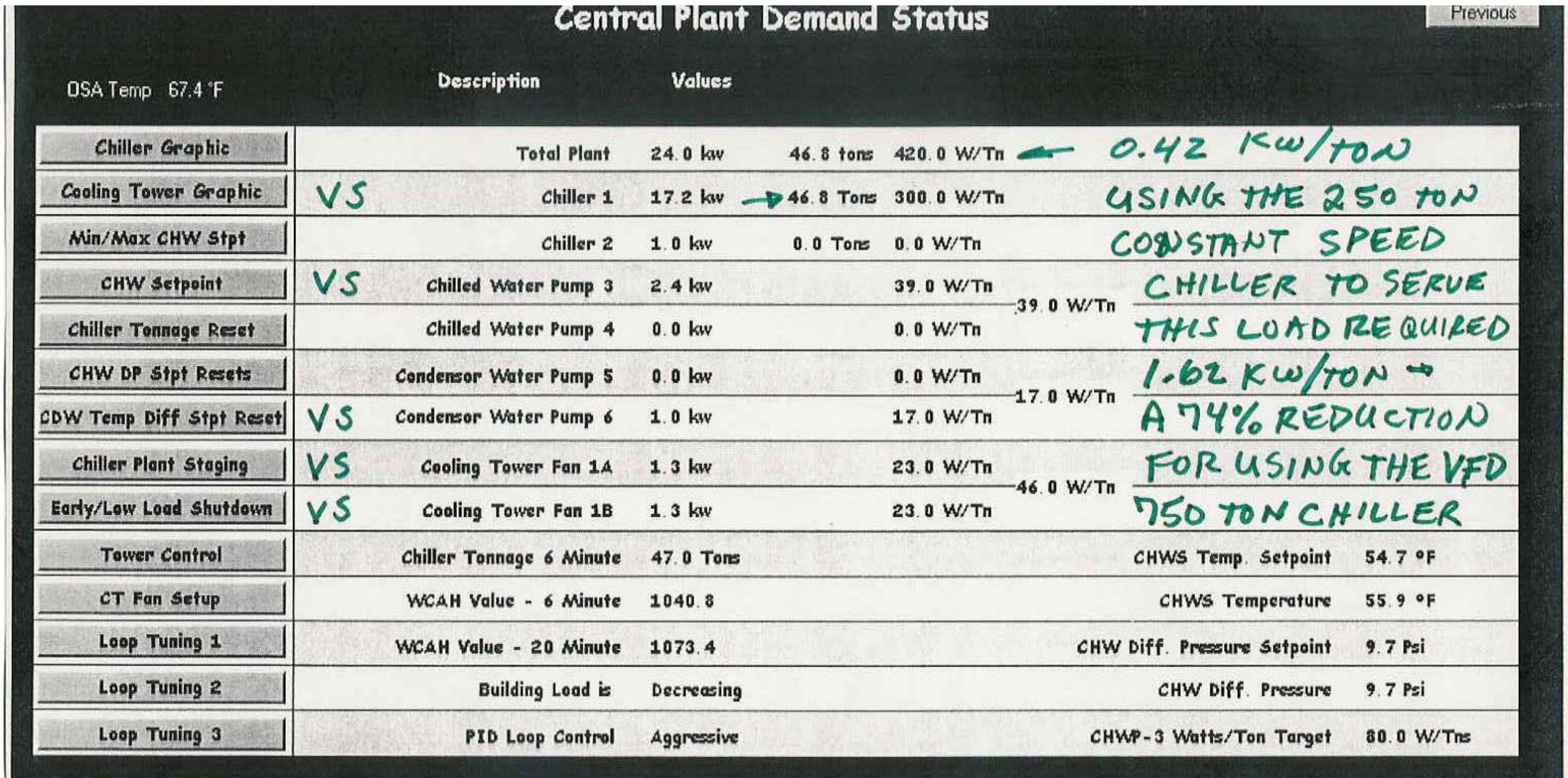
(VSE/LOBOS lite)

Put the VFD on the biggest Chiller you have!

750 ton VFD chiller at 6.3% load (47 tons) – total plant efficiency = 0.42 kW/ton.

250 ton CS chiller at 20% load (49 tons) – total plant efficiency = 1.62 kW/ton.



Central Plant Demand Status

Previous

OSA Temp 67.4 °F

		Description	Values					
Chiller Graphic		Total Plant	24.0 kw	46.8 tons	420.0 W/Tn		0.42 KW/TON	
Cooling Tower Graphic	VS	Chiller 1	17.2 kw	46.8 Tons	300.0 W/Tn		USING THE 250 TON	
Min/Max CHW Stpt		Chiller 2	1.0 kw	0.0 Tons	0.0 W/Tn		CONSTANT SPEED	
CHW Setpoint	VS	Chilled Water Pump 3	2.4 kw		39.0 W/Tn	39.0 W/Tn	CHILLER TO SERVE	
Chiller Tonnage Reset		Chilled Water Pump 4	0.0 kw		0.0 W/Tn		THIS LOAD REQUIRED	
CHW DP Stpt Resets		Condensor Water Pump 5	0.0 kw		0.0 W/Tn	17.0 W/Tn	1.62 KW/TON →	
CDW Temp Diff Stpt Reset	VS	Condensor Water Pump 6	1.0 kw		17.0 W/Tn		A 74% REDUCTION	
Chiller Plant Staging	VS	Cooling Tower Fan 1A	1.3 kw		23.0 W/Tn	46.0 W/Tn	FOR USING THE VFD	
Early/Low Load Shutdown	VS	Cooling Tower Fan 1B	1.3 kw		23.0 W/Tn		750 TON CHILLER	
Tower Control		Chiller Tonnage 6 Minute	47.0 Tons				CHWS Temp. Setpoint	54.7 °F
CT Fan Setup		WCAH Value - 6 Minute	1040.8				CHWS Temperature	55.9 °F
Loop Tuning 1		WCAH Value - 20 Minute	1073.4				CHW Diff. Pressure Setpoint	9.7 Psi
Loop Tuning 2		Building Load is	Decreasing				CHW Diff. Pressure	9.7 Psi
Loop Tuning 3		PID Loop Control	Aggressive				CHWP-3 Watts/Ton Target	80.0 W/Tns

Summary

- **It is possible to reduce cooling system related energy consumption and GHG Emissions by 30% to 50% or more, while improving comfort conditions.**
- **The Optimization Routines must look at 100% of the HVAC energy consuming devices, and the comfort conditions of the facility, not just the chiller plant.**
- **The Load Based Optimization System (LOBOS) has been the key to many successful projects.**
- **Operating Engineers are the key to long term success. If they don't understand it, or they can't "tweak" it to suit their facility, it won't work for long!**
- **The system must be easy to adjust to meet their needs.**

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Summary, continued

- **It is not that tough to save a substantial amount of energy and reduce GHG emissions, while at the same time improving comfort conditions.**
- **Proper equipment and system sizing and selection should be based on the lifecycle cost of the facility, reducing biological issues and promoting tenant retention.**
- **The bottom line will be up to your operating staff, if they can tune the control system to suit the specific needs of the loads being served, the system has a far greater chance of success, than if the system cannot be easily tuned.**

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